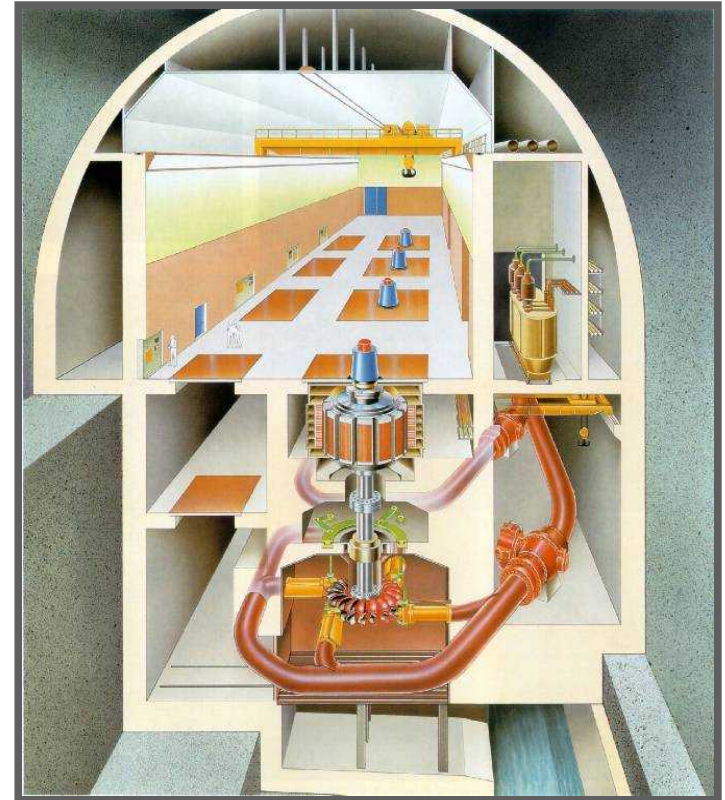
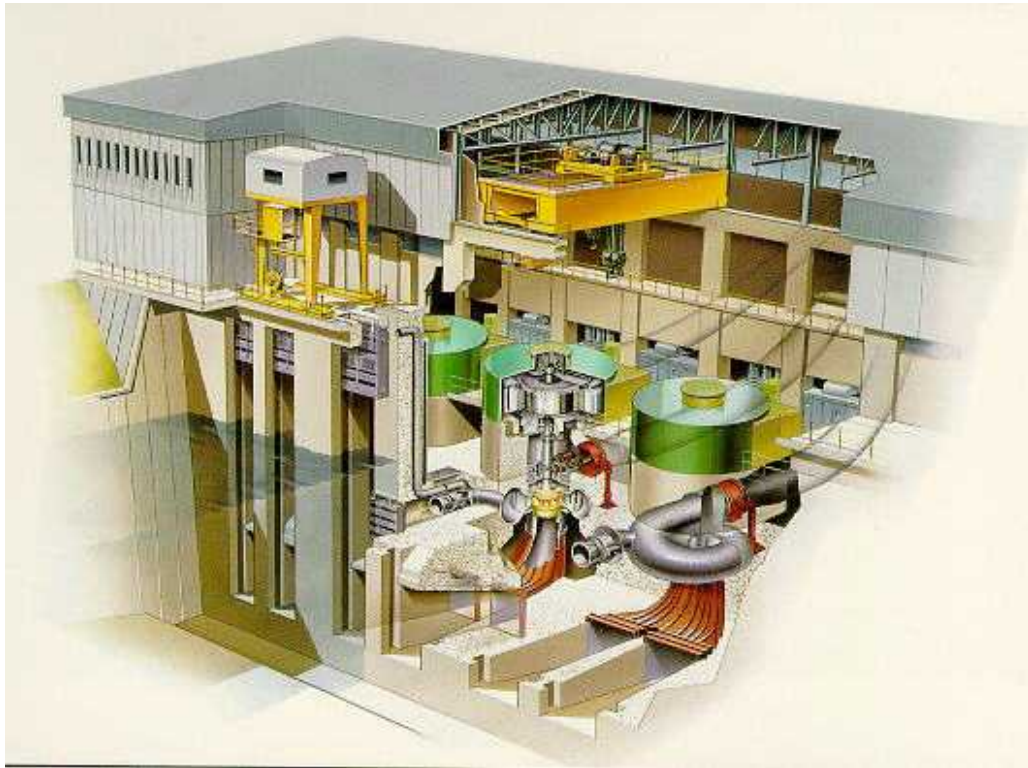
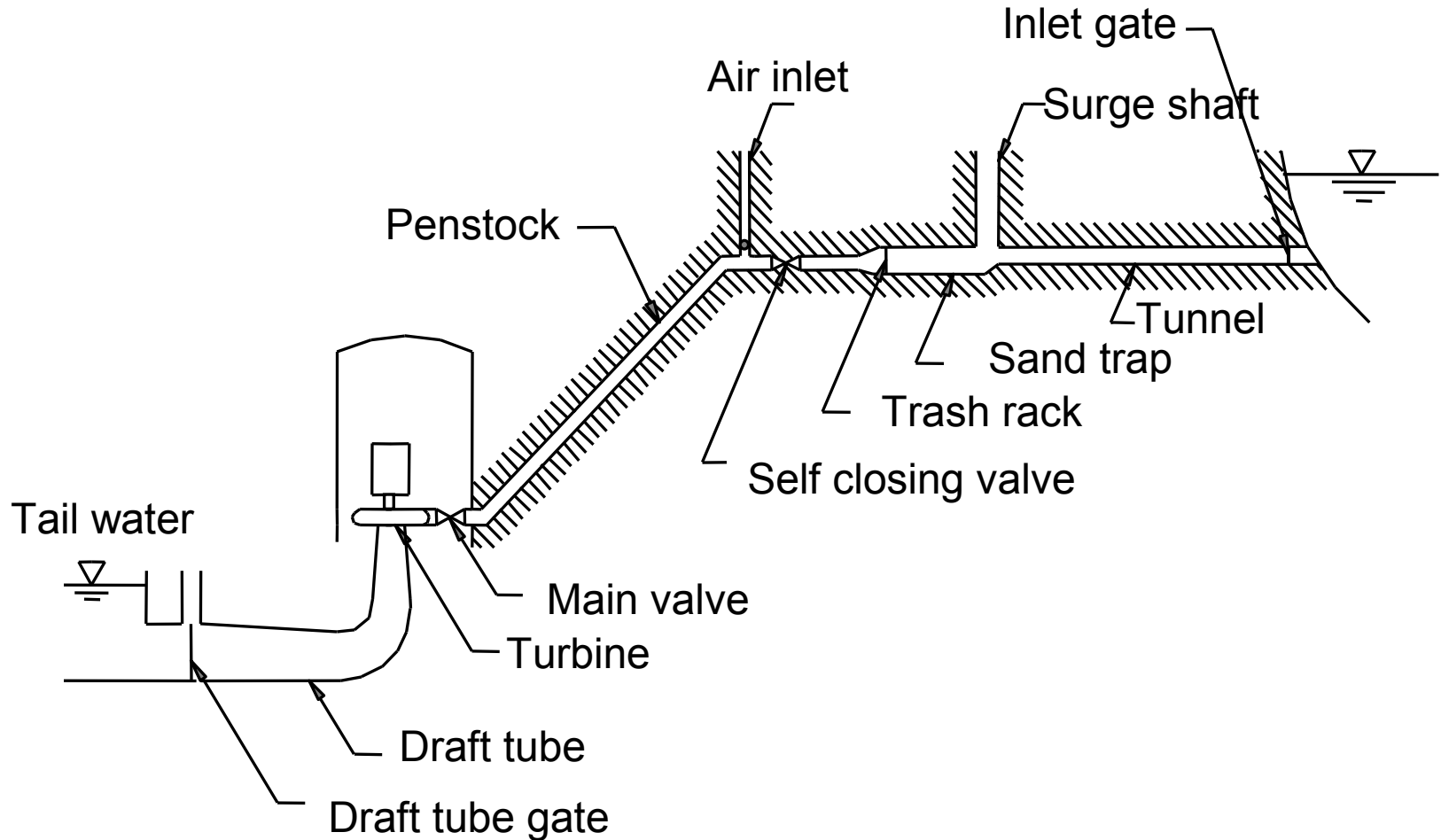


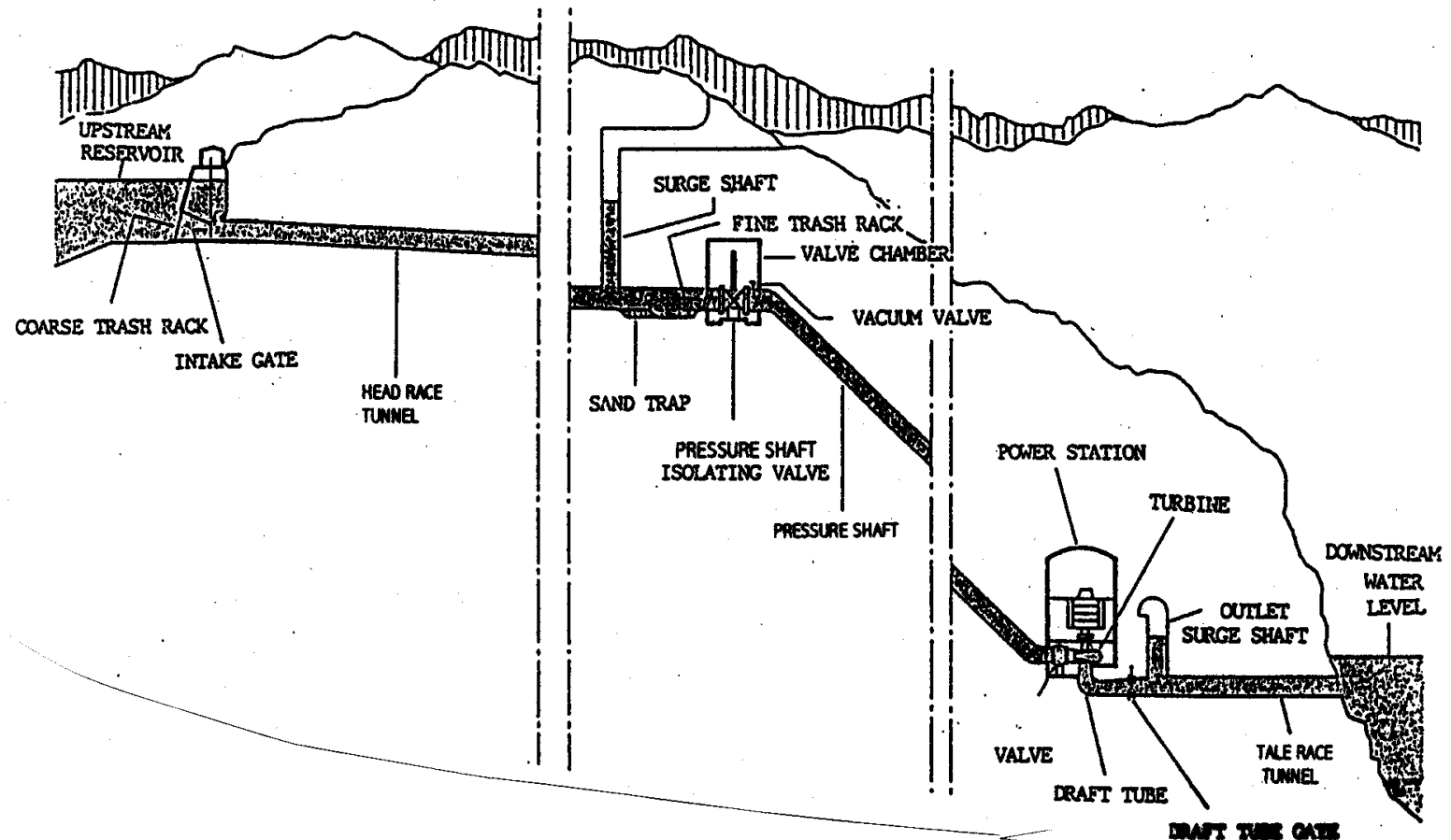
Hydro power plants



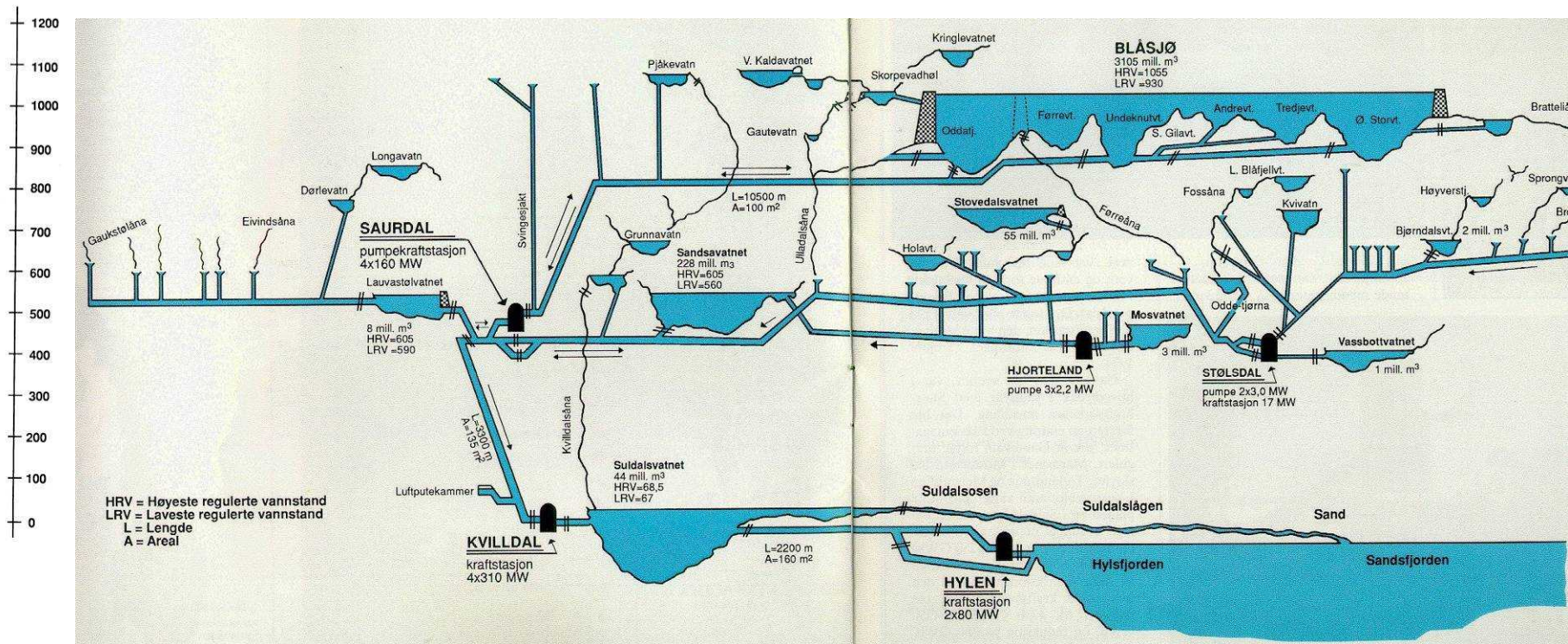
Hydro power plants



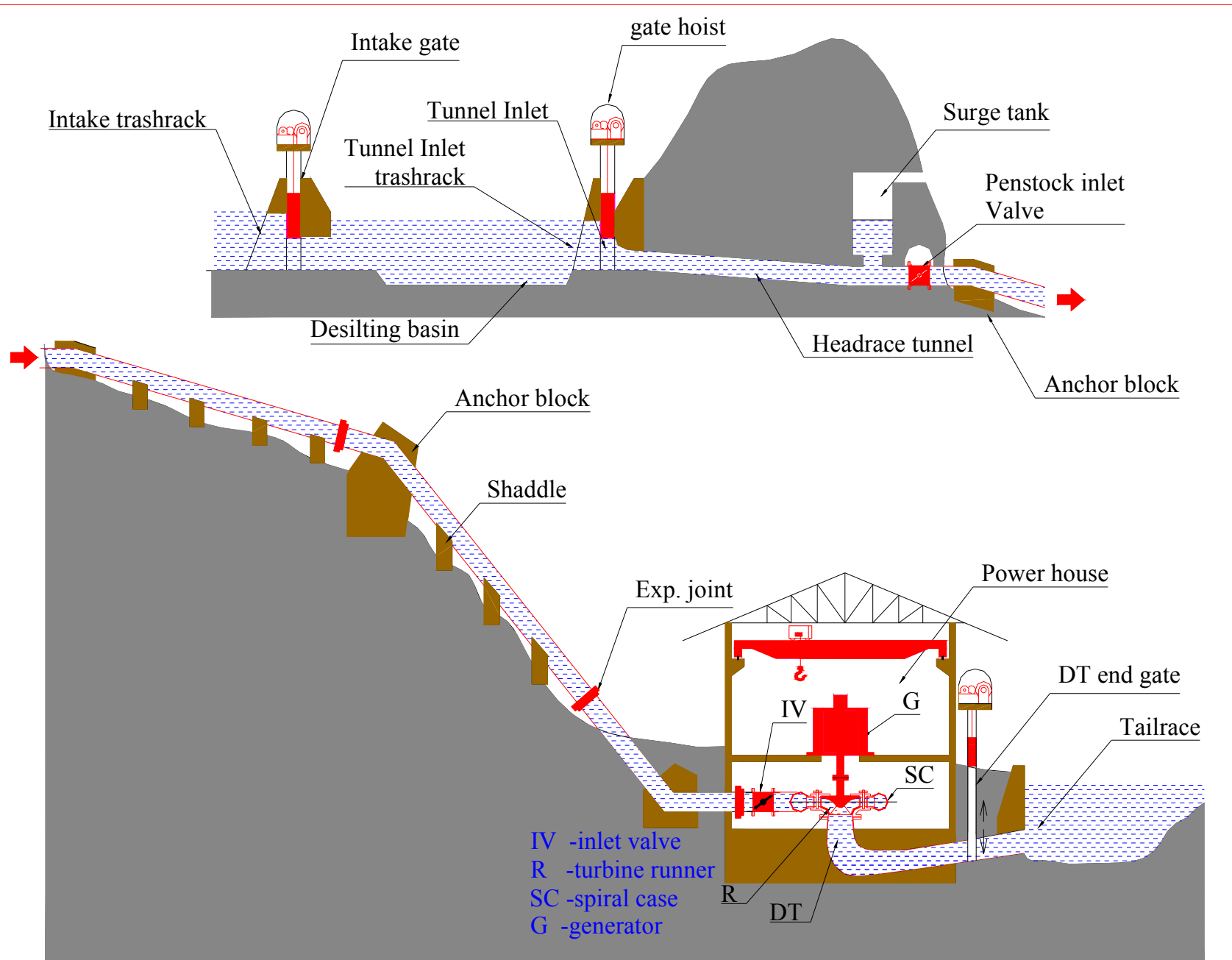
The principle the water conduits of a traditional high head power plant



Ulla- Førre

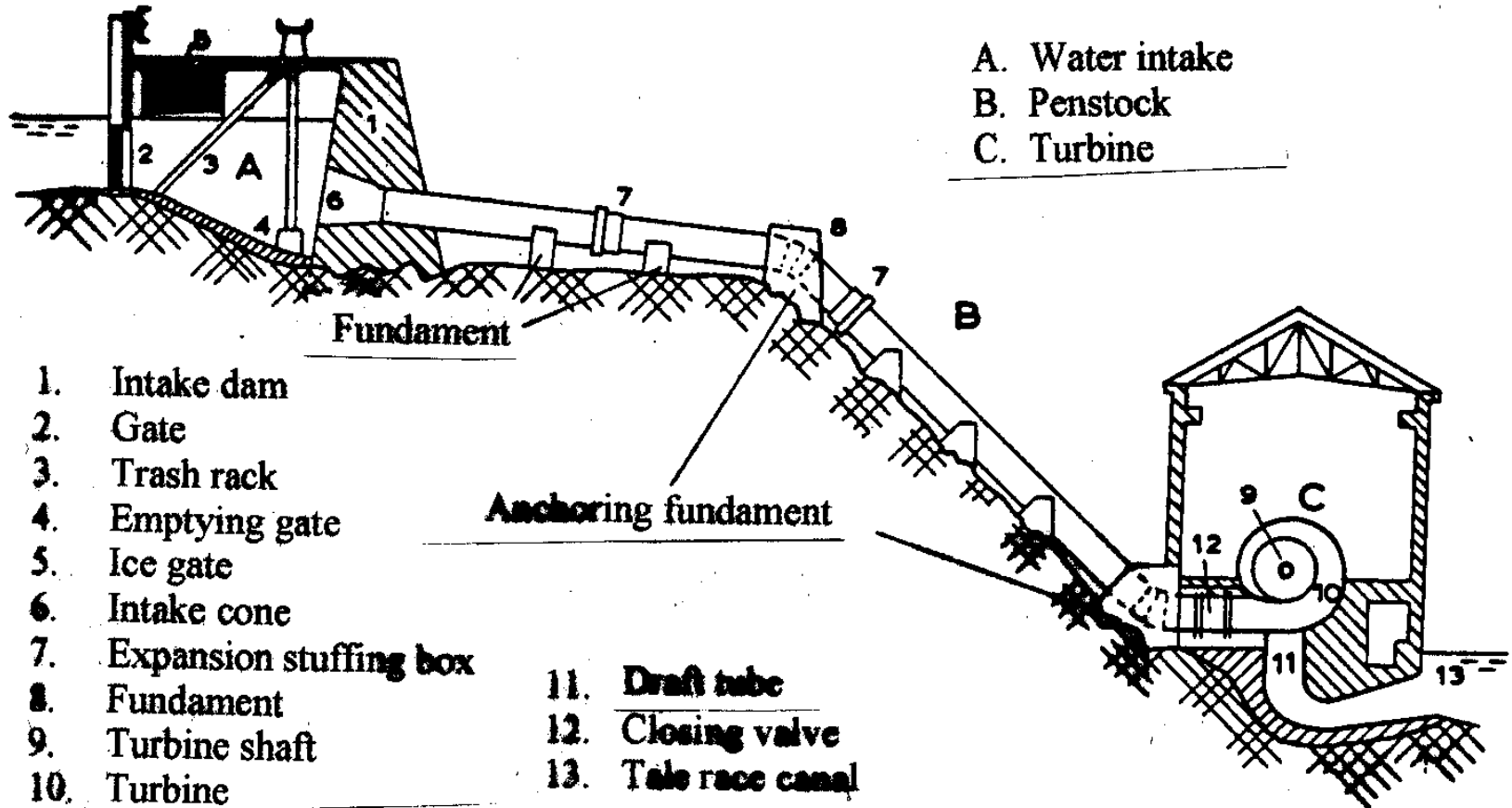


Original figur ved Statkraft Vestlandsverkene

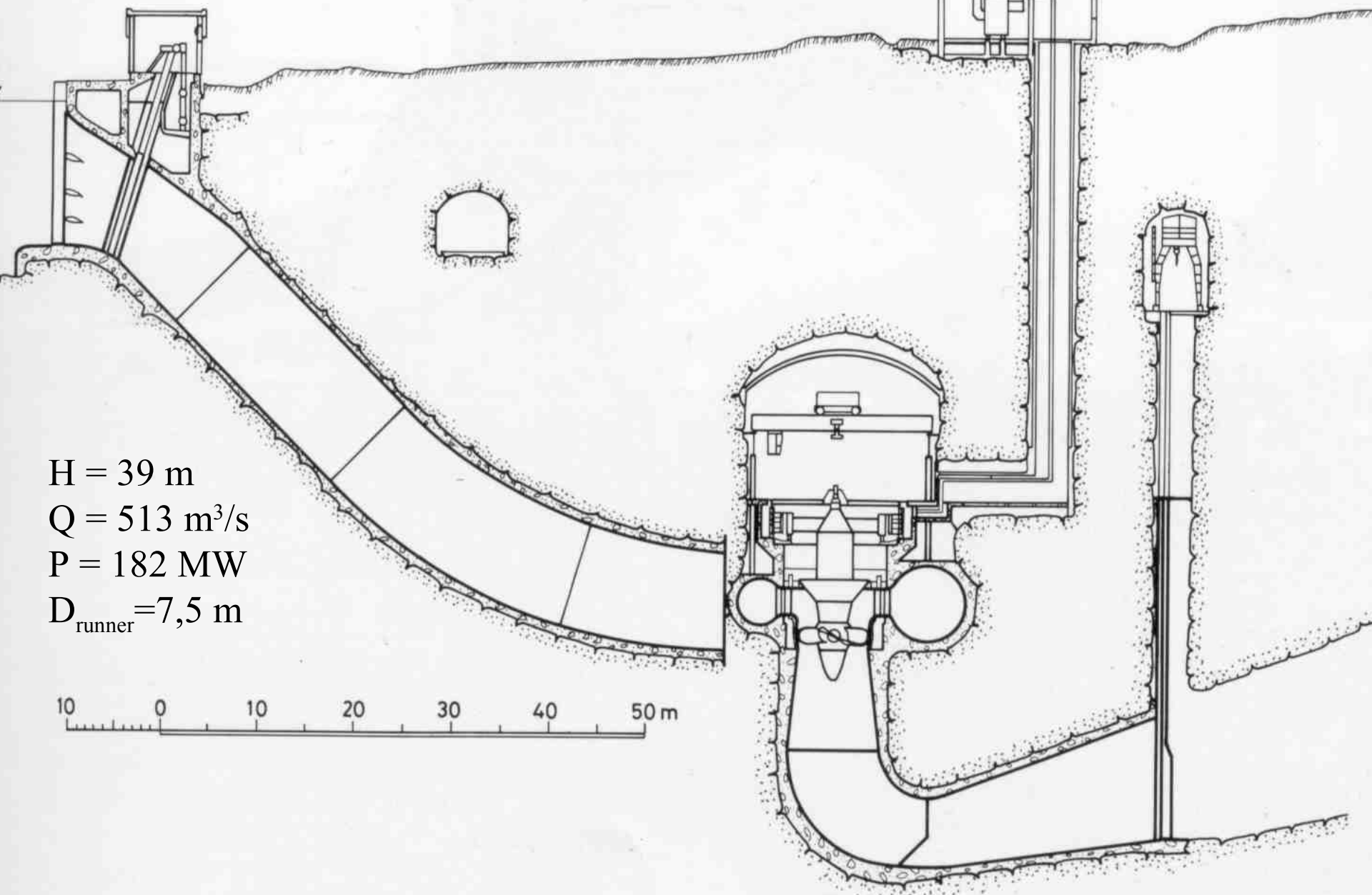


Typical Power House with Francis Turbine

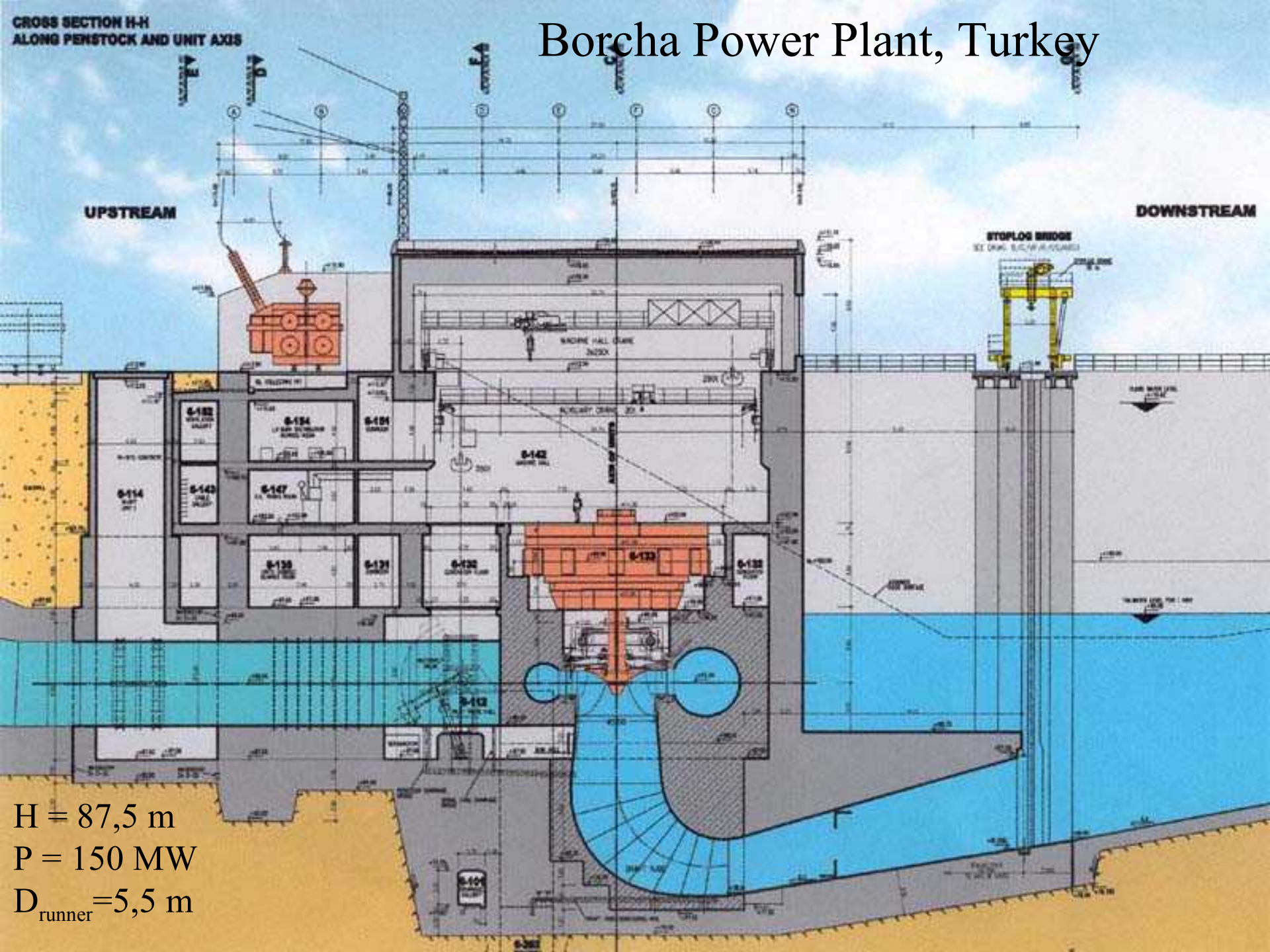
Arrangement of a small hydropower plant



Ligga Power Plant, Norrbotten, Sweden



Borçea Power Plant, Turkey



Water intake

- Dam
- Coarse trash rack
- Intake gate
- Sediment settling basement

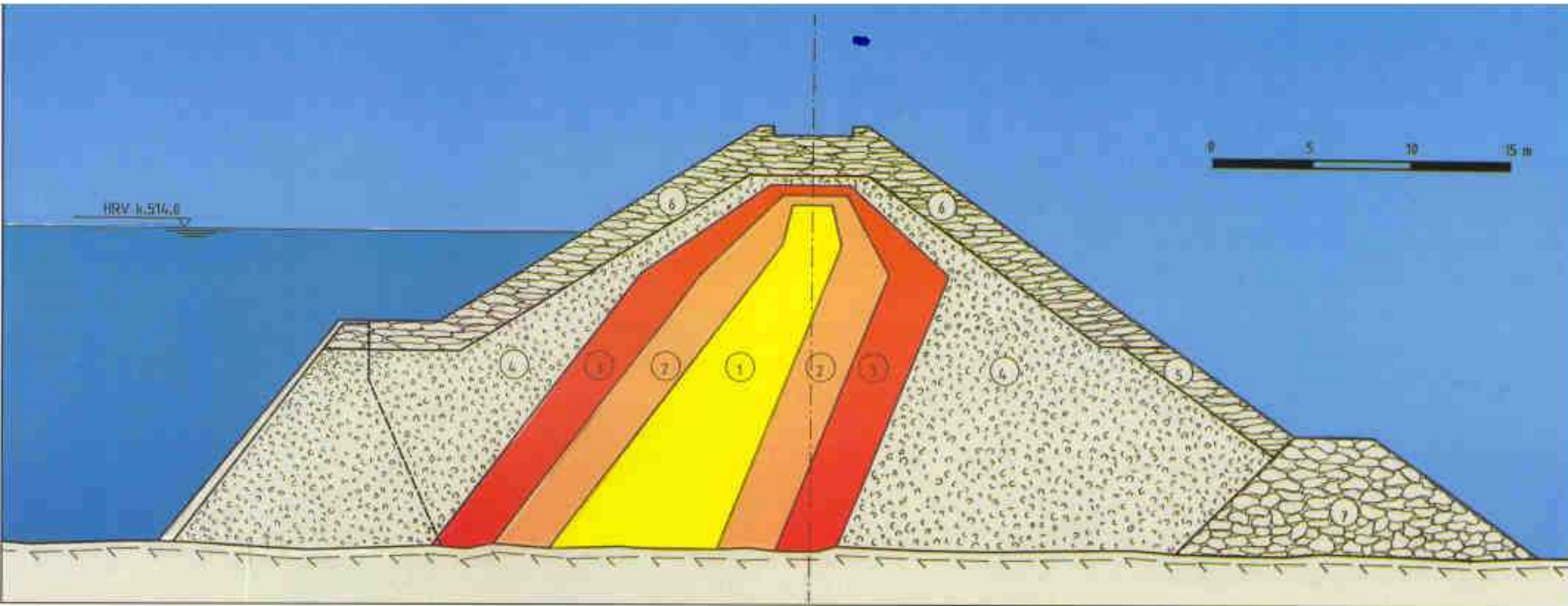


Dams

- Rockfill dams
- Pilar og platedammer
- Hvelvdammer



Rock-fill dams



- | | |
|---------------------|---|
| 1. Core | Moraine, crushed soft rock, concrete, asphalt |
| 2. Filter zone | Sandy gravel |
| 3. Transition zone | Fine blasted rock |
| 4. Supporting shell | Blasted rock |

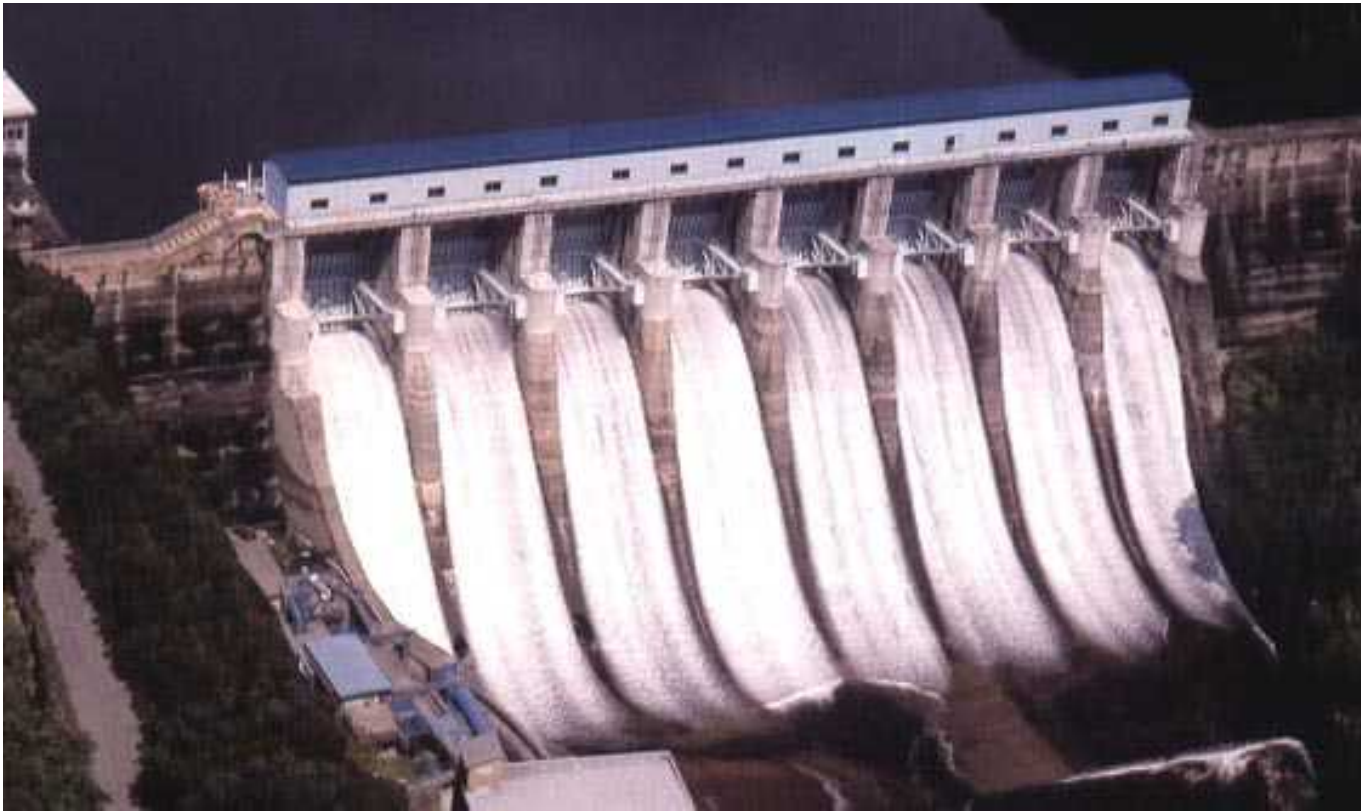
Slab concrete dam



Arc dam



Gates in Hydro Power Plants



Types of Gates

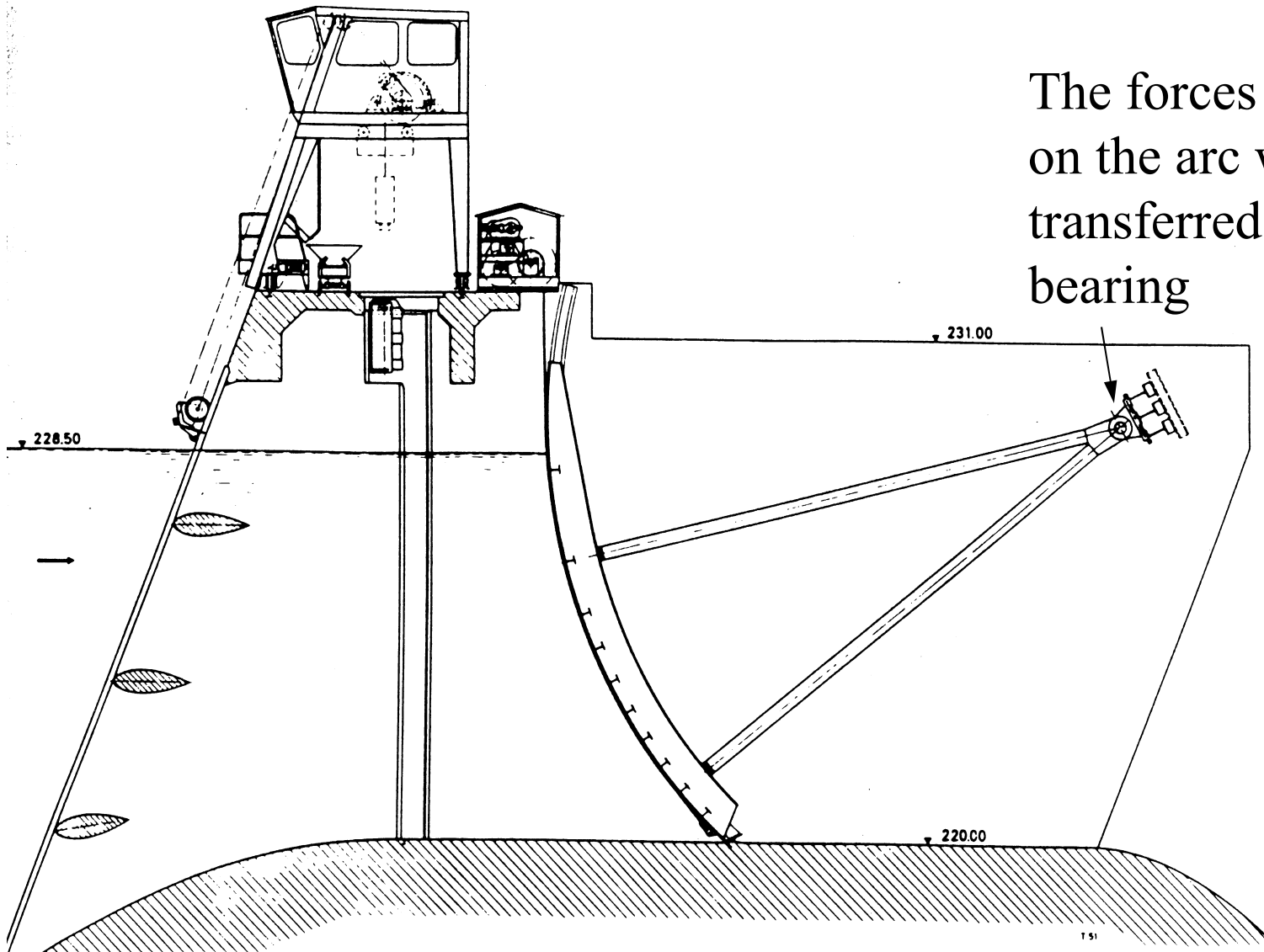
- Radial Gates
- Wheel Gates
- Slide Gates
- Flap Gates
- Rubber Gates

Radial Gates at Älvkarleby, Sweden



Radial Gate

The forces acting on the arc will be transferred to the bearing



Slide Gate

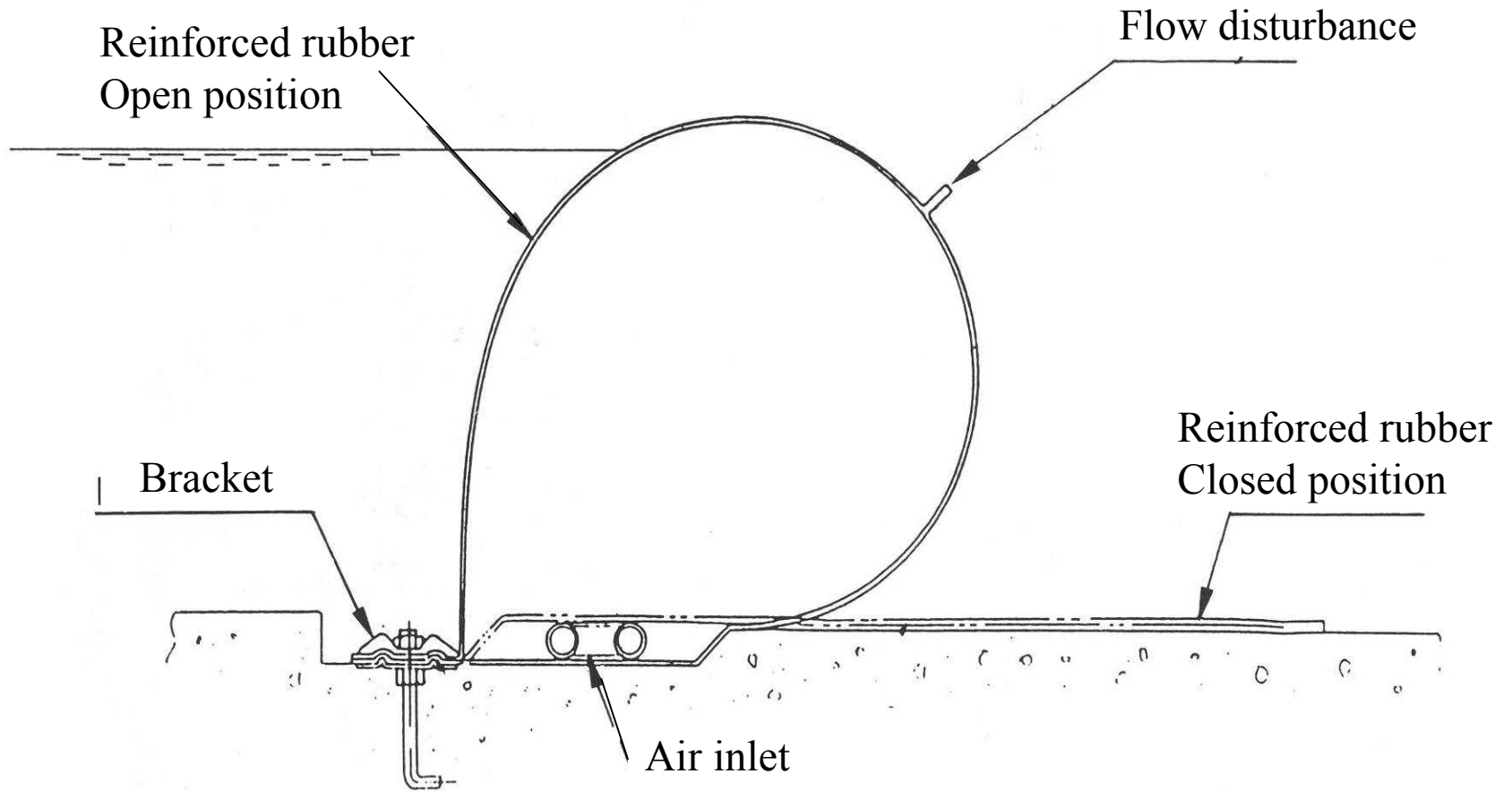


Jhimruk Power Plant, Nepal

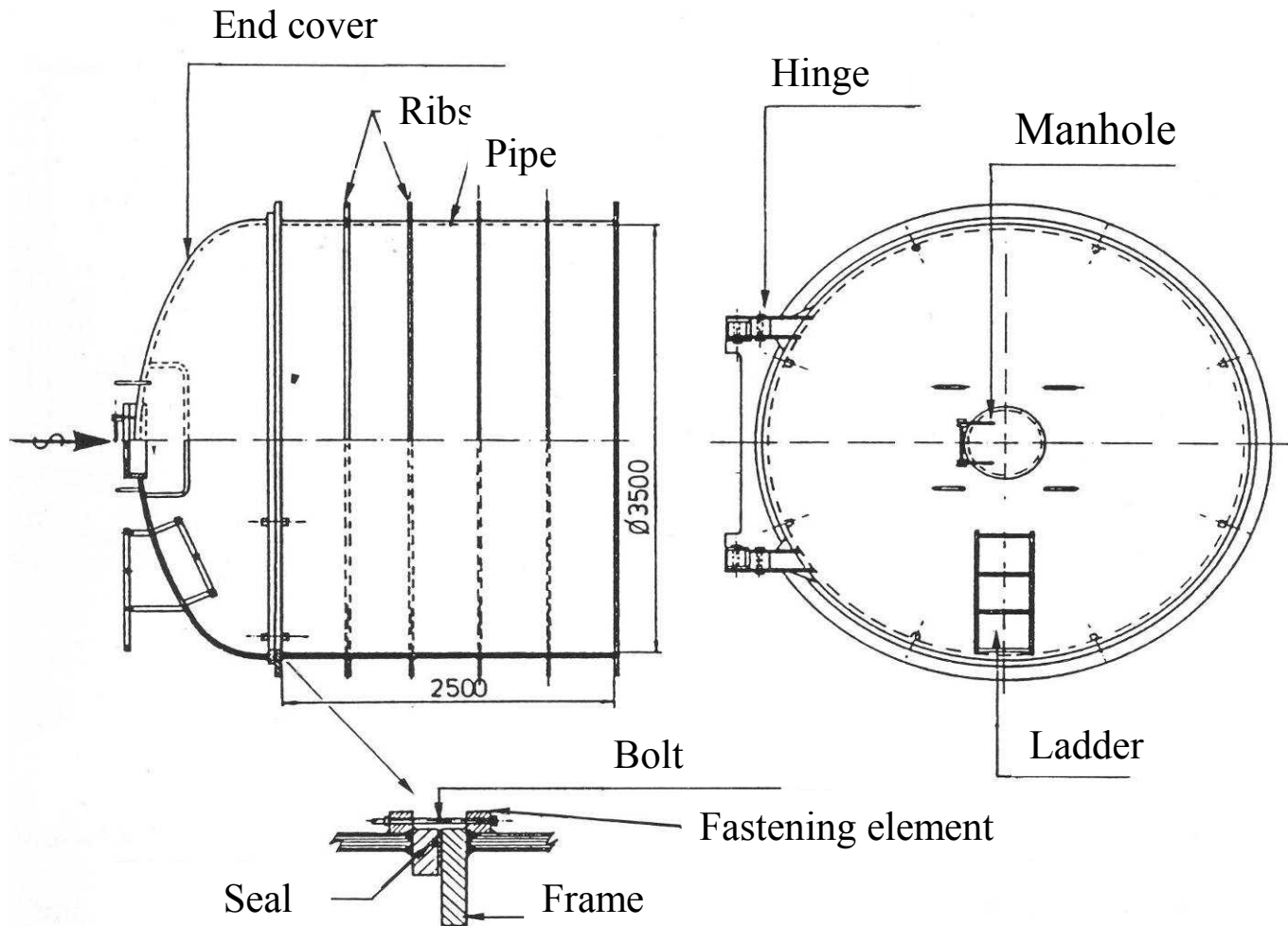
Flap Gate



Rubber gate



Circular gate



Circular gate



Jhimruk Power Plant, Nepal

Trash Racks



Panauti Power Plant, Nepal

Theun Hinboun Power Plant

Laos



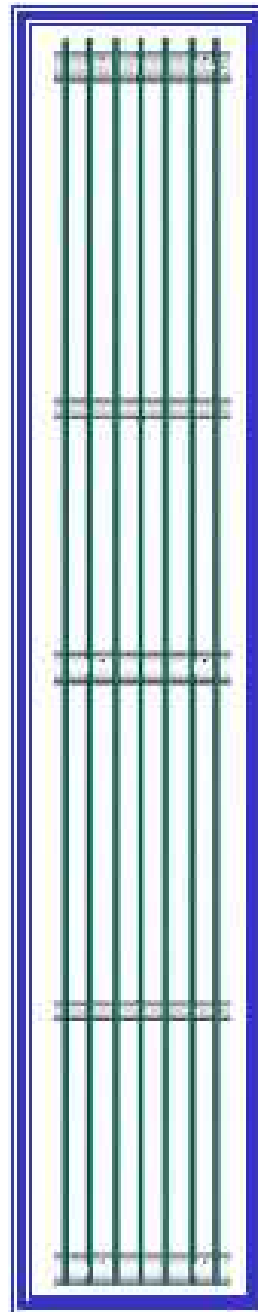
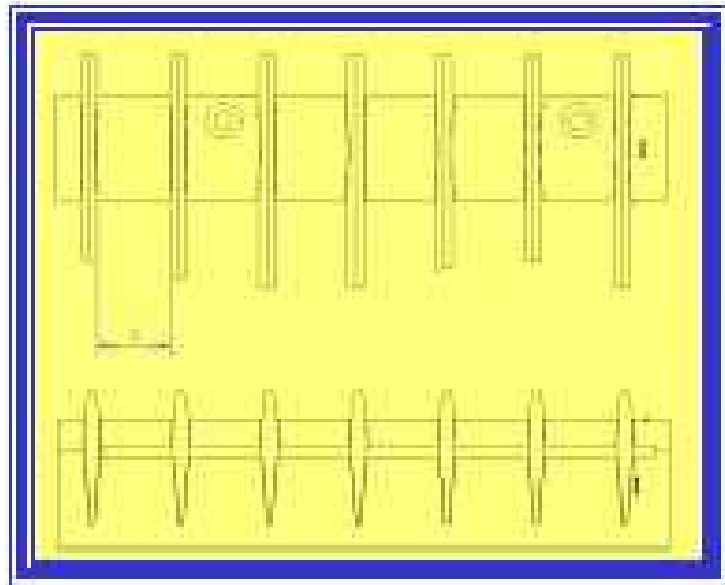
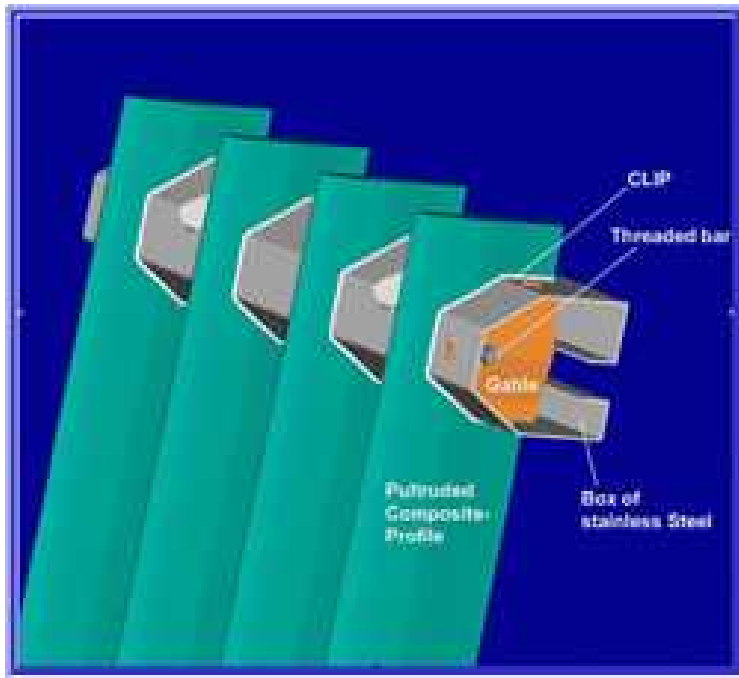
Gravfoss

Power Plant
Norway

Trash Rack size:
Width: 12 meter
Height: 13 meter

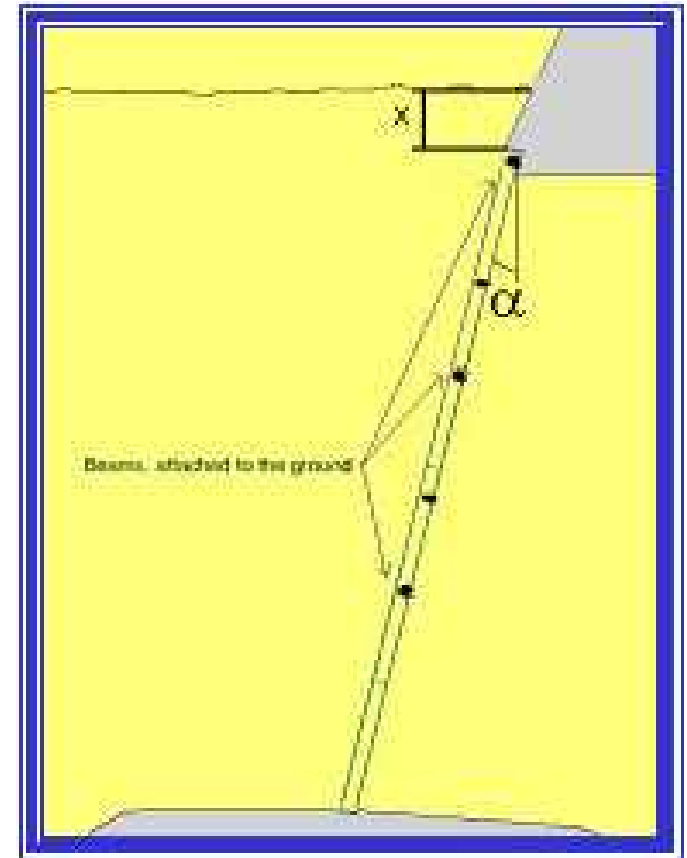
Stainless Steel



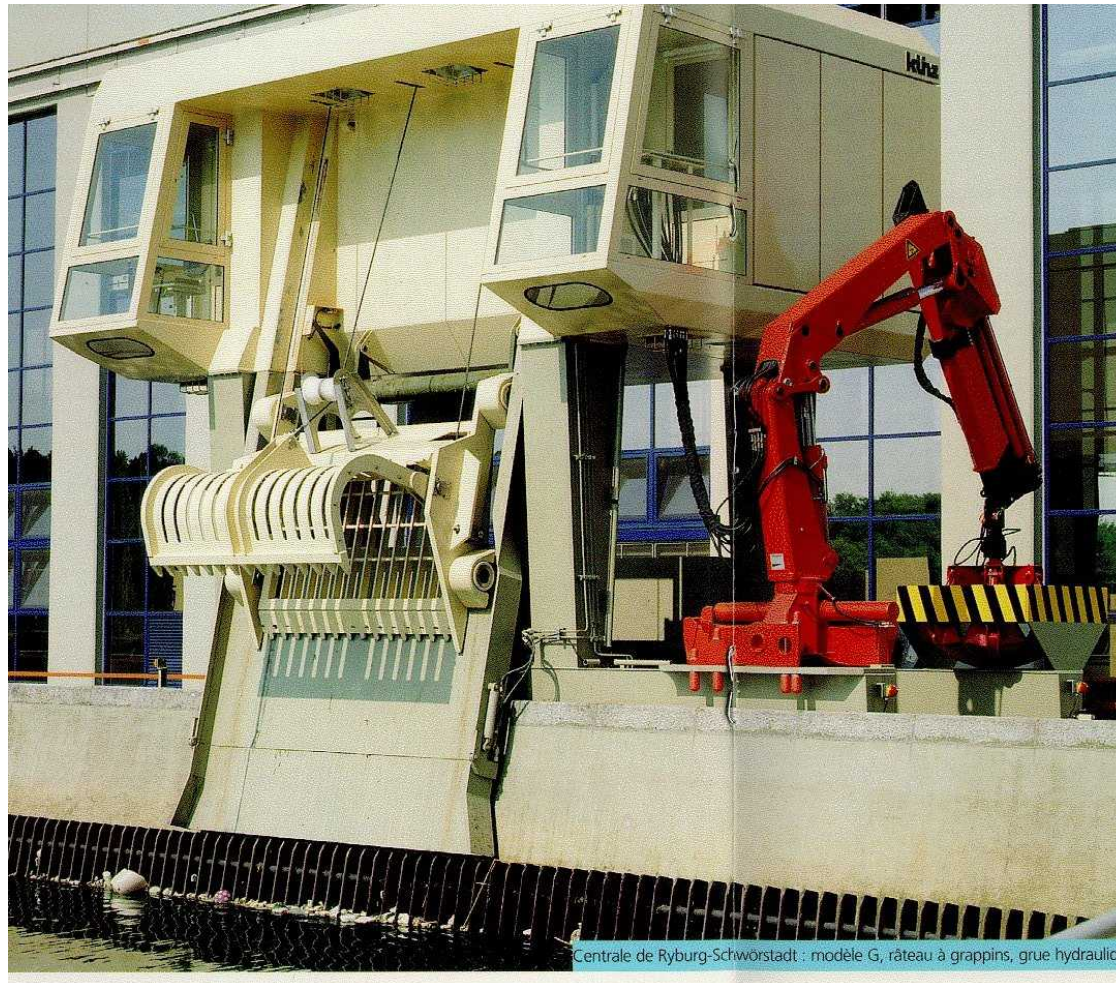


CompRack

Trash Rack delivered by VA-Tech



Cleaning the trash rack



Centrale de Ryburg-Schwörstadt : modèle G, râseau à grappins, grue hydraulique

Pipes

- Materials
- Calculation of the change of length due to the change of the temperature
- Calculation of the head loss
- Calculation of maximum pressure
 - Static pressure
 - Water hammer
- Calculation of the pipe thickness
- Calculation of the economical correct diameter
- Calculation of the forces acting on the anchors

Materials

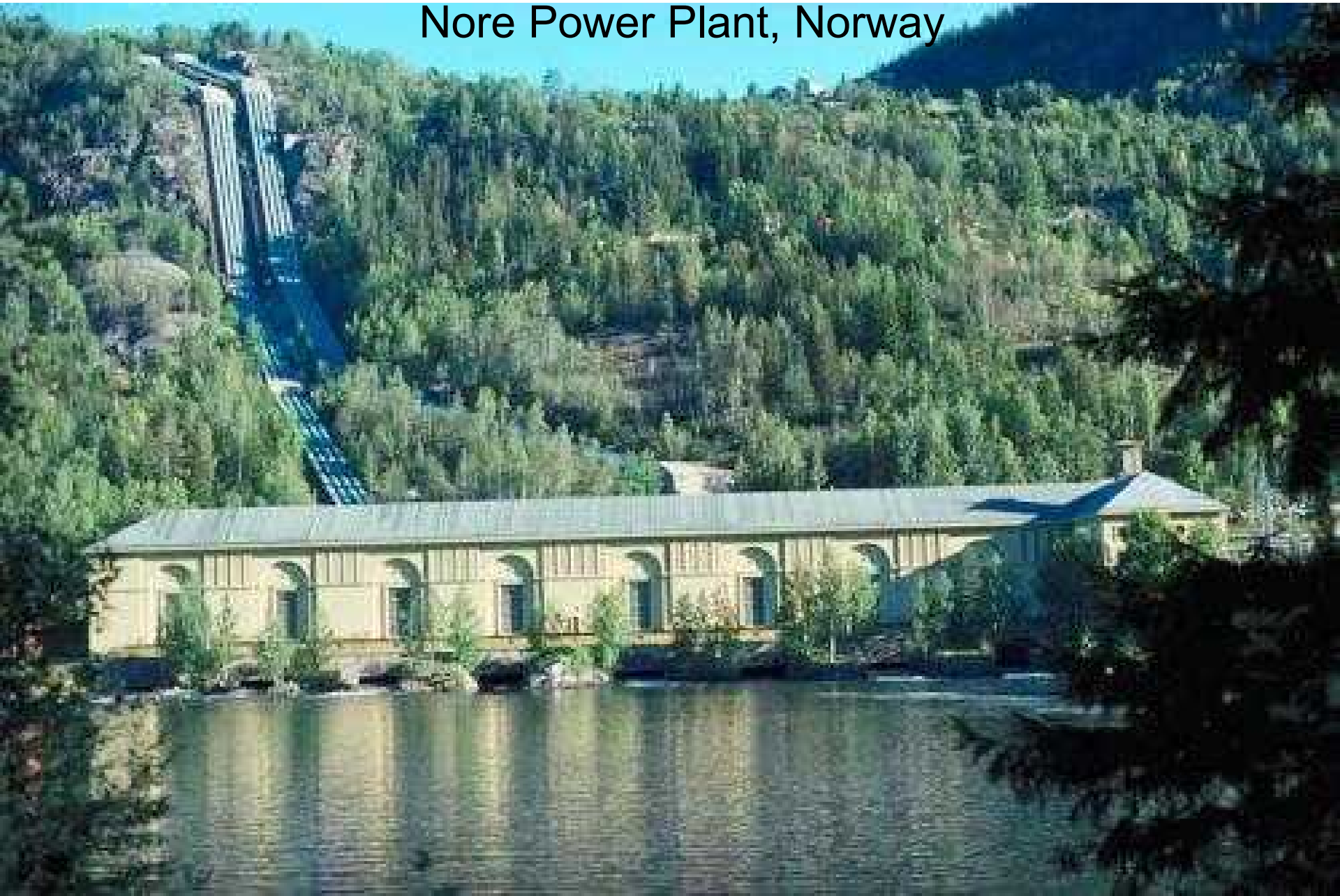
- Steel
- Polyethylene, PE
- Glass-fibre reinforced Unsaturated Polyesterplastic , GUP
- Wood
- Concrete

Materials

Material	Max. Diameter	Max. Pressure	Max. Stresses
	[m]	[m]	[MPa]
Steel, St.37			150
Steel, St.42			190
Steel, St.52			206
PE	~ 1,0	160	5
GUP	2,4 Max. p = 160 m.	320 Max. D: 1,4 m.	
Wood	~5	80	
Concrete	~5	~ 400	

Steel pipes in penstock

Nore Power Plant, Norway



GUP-Pipe

Raubergfossen Power Plant, Norway



Wood Pipes



Breivikbotn Power Plant, Norway



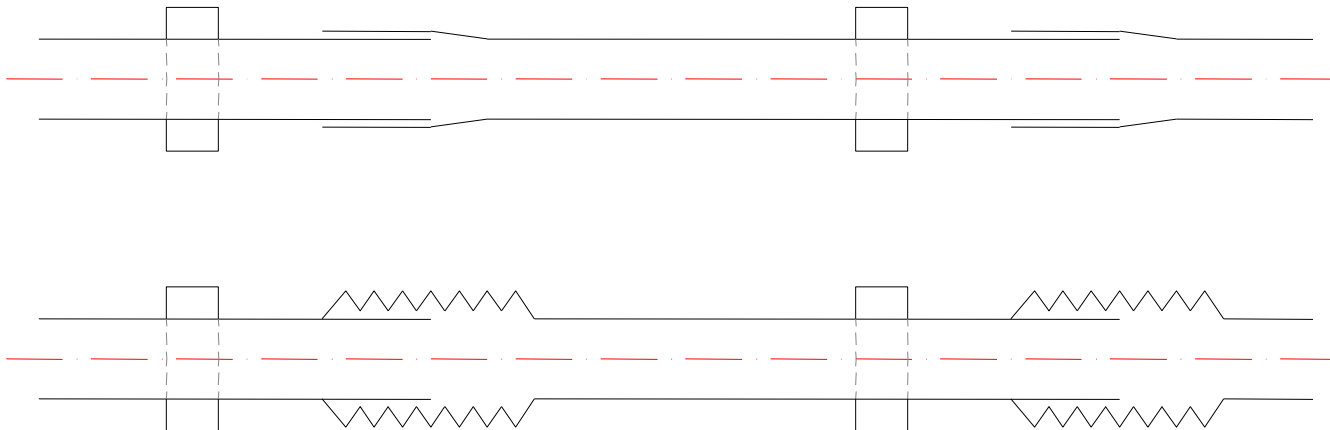
Øvre Porsa Power Plant, Norway

Calculation of the change of length due to the change of the temperature

$$\Delta L = \alpha \cdot \Delta T \cdot L$$

Where:

ΔL = Change of length [m]
 L = Length [m]
 α = Coefficient of thermal expansion [m/°C m]
 ΔT = Change of temperature [°C]



Calculation of the head loss

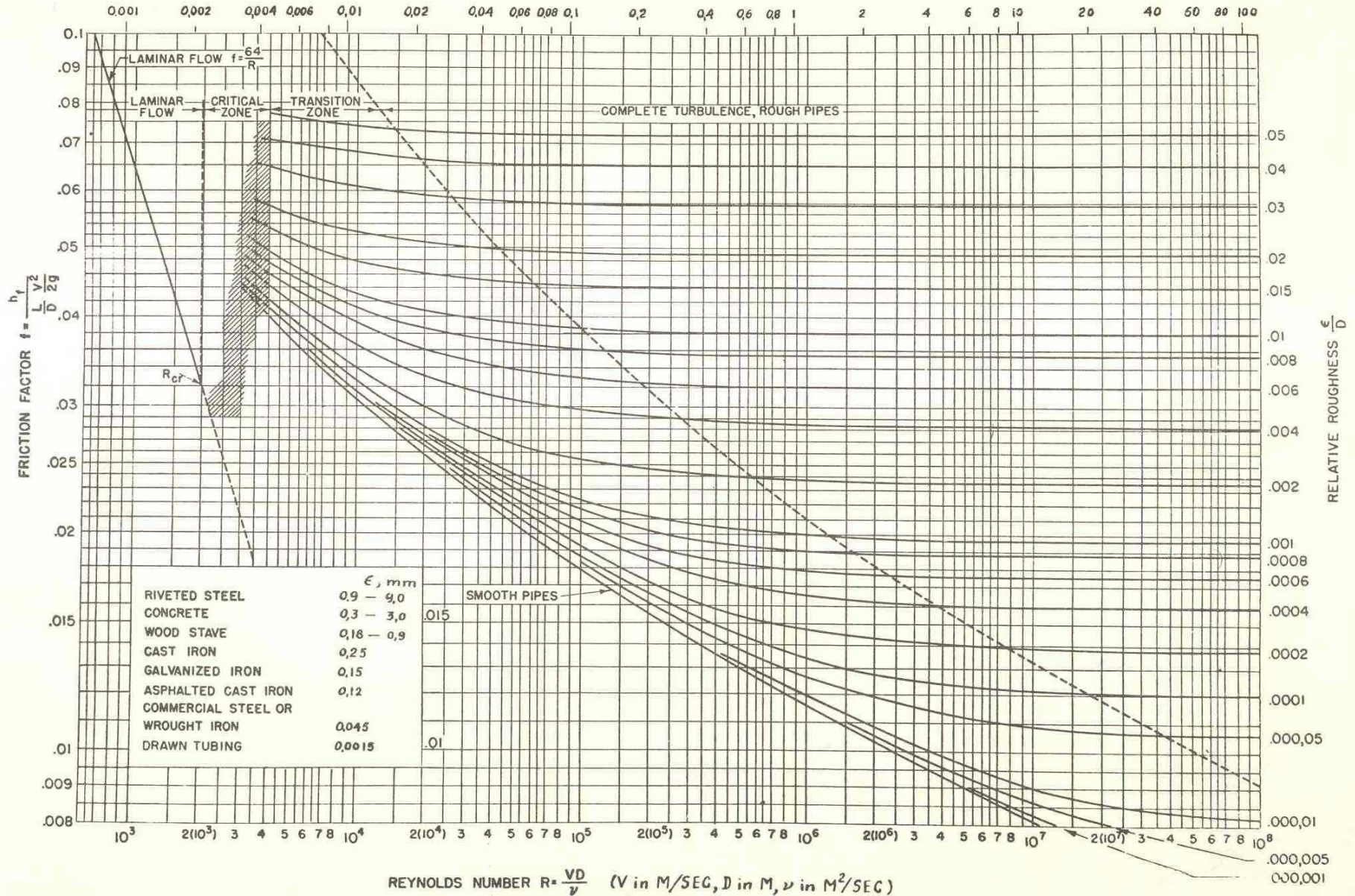
$$h_f = f \cdot \frac{L}{D} \cdot \frac{c^2}{2 \cdot g}$$

Where:

h_f	= Head loss	[m]
f	= Friction factor	[-]
L	= Length of pipe	[m]
D	= Diameter of the pipe	[m]
c	= Water velocity	[m/s]
g	= Gravity	[m/s ²]

MOODY DIAGRAM

VALUES OF (VD^*) FOR WATER AT 15.5°C (VELOCITY IN M/SEC * DIAMETER IN M)



Example

Calculation of the head loss

Power Plant data:

H	=	100 m	Head
Q	=	10 m ³ /s	Flow Rate
L	=	1000 m	Length of pipe
D	=	2,0 m	Diameter of the pipe
The pipe material is steel			

$$h_f = f \cdot \frac{L}{D} \cdot \frac{c^2}{2 \cdot g}$$

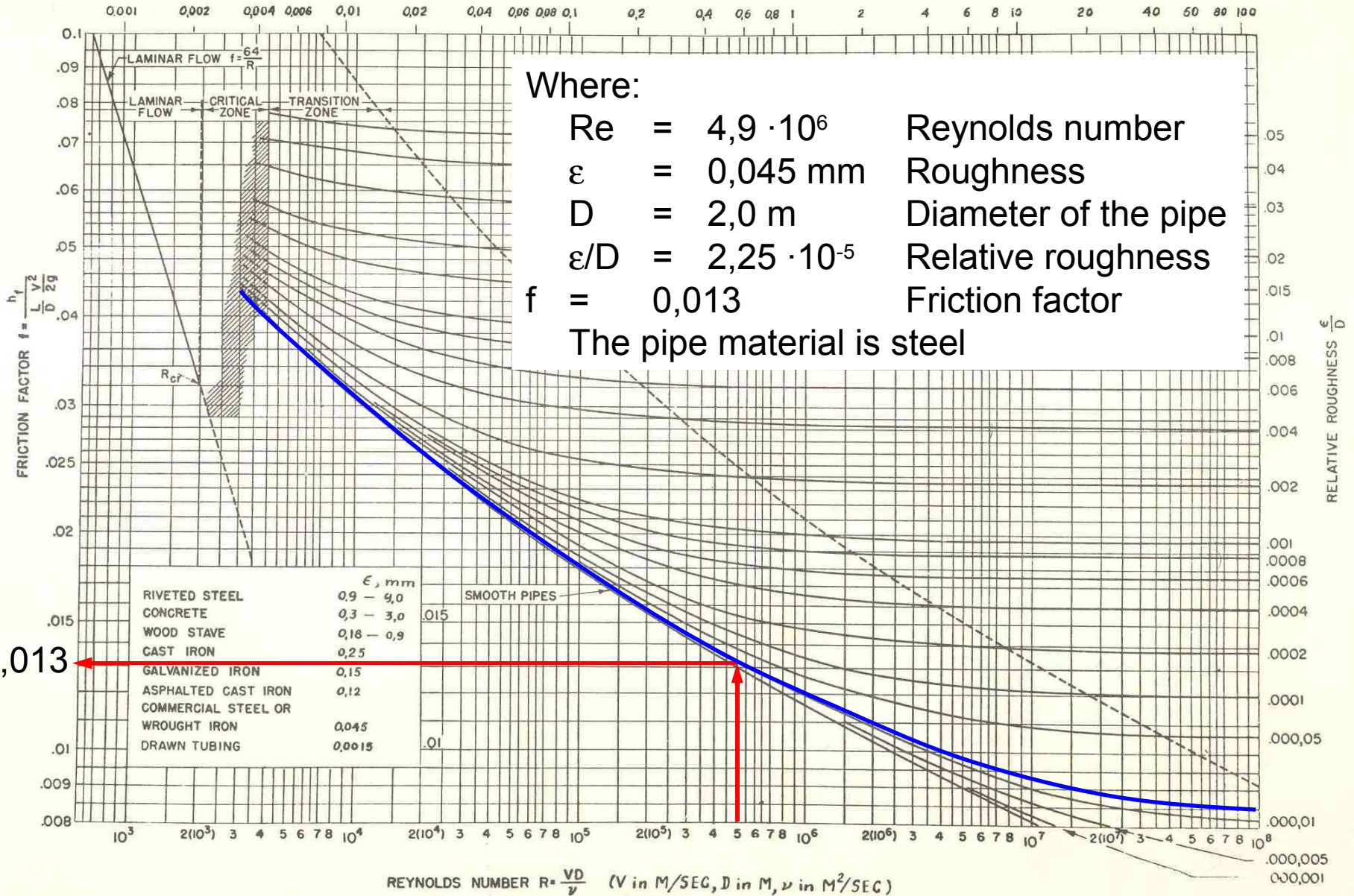
$$Re = \frac{c \cdot D}{\nu}$$

Where:

c	=	3,2 m/s	Water velocity
ν	=	1,308·10 ⁻⁶ m ² /s	Kinetic viscosity
Re	=	4,9 · 10 ⁶	Reynolds number

MOODY DIAGRAM

VALUES OF (VD^*) FOR WATER AT 15,5°C (VELOCITY IN M/SEC * DIAMETER IN M)



Example

Calculation of the head loss

Power Plant data:

H = 100 m Head
Q = 10 m³/s Flow Rate
L = 1000 m Length of pipe
D = 2,0 m Diameter of the pipe
The pipe material is steel

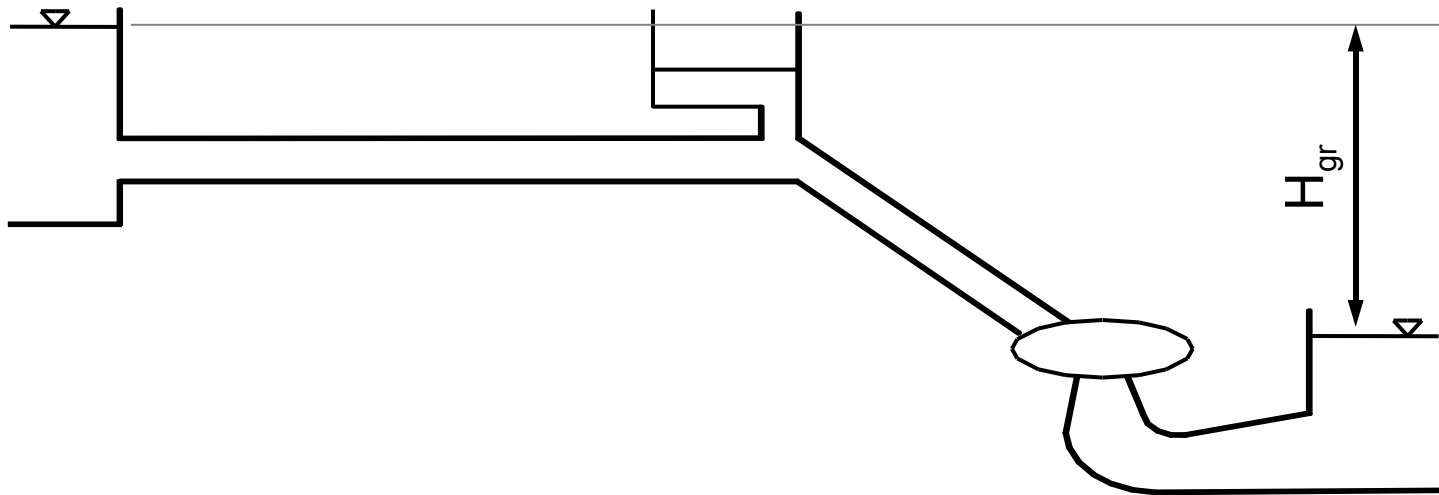
$$h_f = f \cdot \frac{L}{D} \cdot \frac{c^2}{2 \cdot g} = 0,013 \cdot \frac{1000}{2} \cdot \frac{3,2^2}{2 \cdot 9,82} = 3,4 \text{ m}$$

Where:

f = 0,013 Friction factor
c = 3,2 m/s Water velocity
g = 9,82 m/s² Gravity

Calculation of maximum pressure

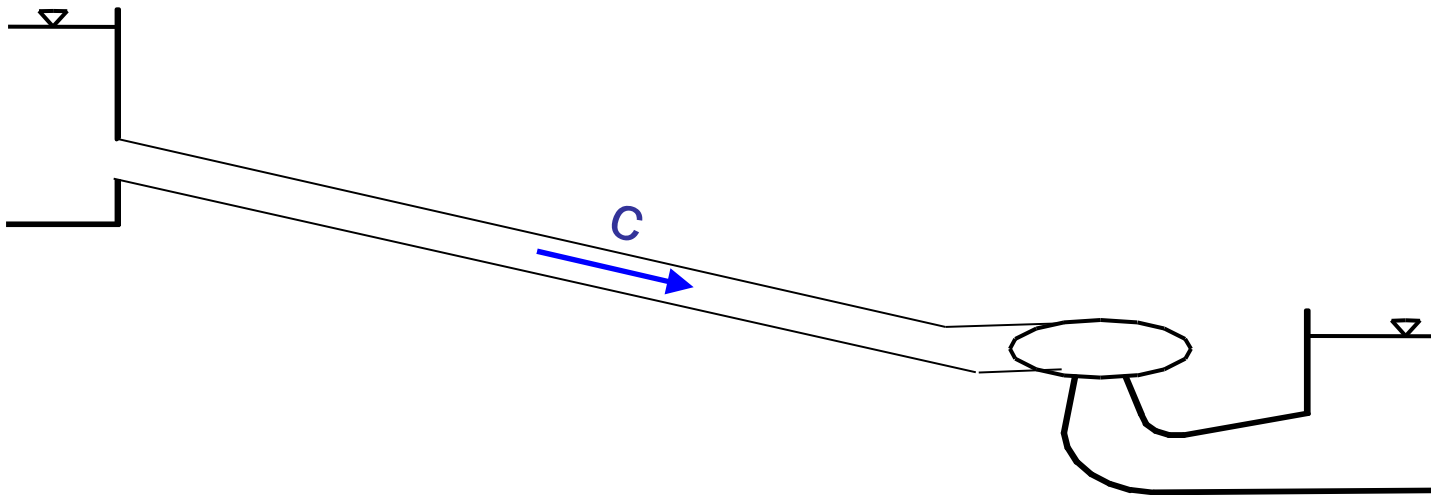
- Static head, H_{gr} (Gross Head)
- Water hammer, Δh_{wh}
- Deflection between pipe supports
- Friction in the axial direction



Maximum pressure rise due to the Water Hammer

$$\Delta h_{wh} = \frac{a \cdot c_{max}}{g} \quad \text{IF } T_c \ll \frac{2 \cdot L}{a} \quad \text{Jowkowsky}$$

Δh_{wh}	= Pressure rise due to water hammer	[mWC]
a	= Speed of sound in the penstock	[m/s]
c_{max}	= maximum velocity	[m/s]
g	= gravity	[m/s ²]



Example

Jowkowsky

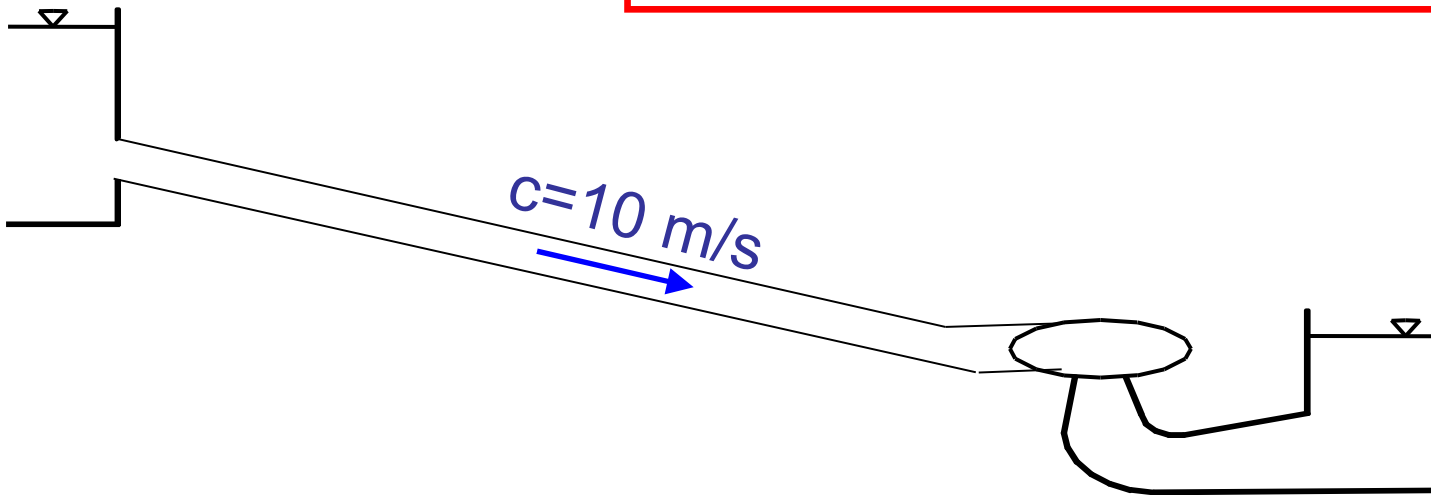
$$a = 1000 \text{ [m/s]}$$

$$c_{\max} = 10 \text{ [m/s]}$$

$$g = 9,81 \text{ [m/s}^2\text{]}$$

$$T_C \ll \frac{2 \cdot L}{a}$$

$$\Delta h_{\text{wh}} = \frac{a \cdot c_{\max}}{g} = 1020 \text{ m}$$

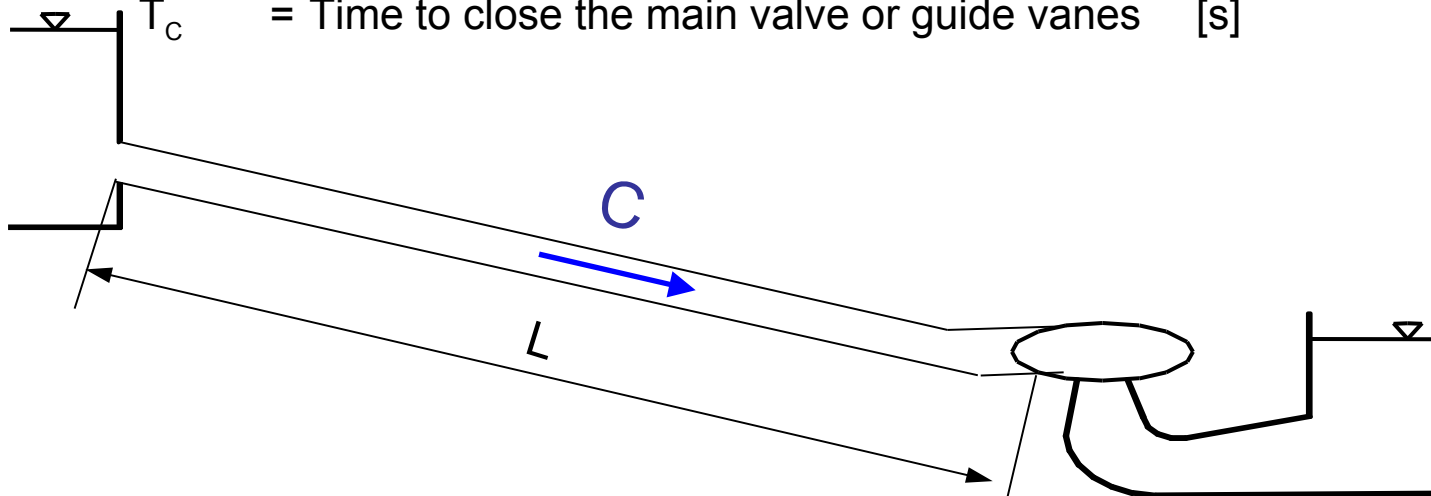


Maximum pressure rise due to the Water Hammer

$$\Delta h_{wh} = \frac{a \cdot c_{max}}{g} \cdot \frac{2 \cdot L / a}{T_C} = \frac{c_{max} \cdot 2 \cdot L}{g \cdot T_C} \quad \text{IF } T_C \geq \frac{2 \cdot L}{a}$$

Where:

Δh_{wh}	= Pressure rise due to water hammer	[mWC]
a	= Speed of sound in the penstock	[m/s]
c_{max}	= maximum velocity	[m/s]
g	= gravity	[m/s ²]
L	= Length	[m]
T_C	= Time to close the main valve or guide vanes	[s]



Example

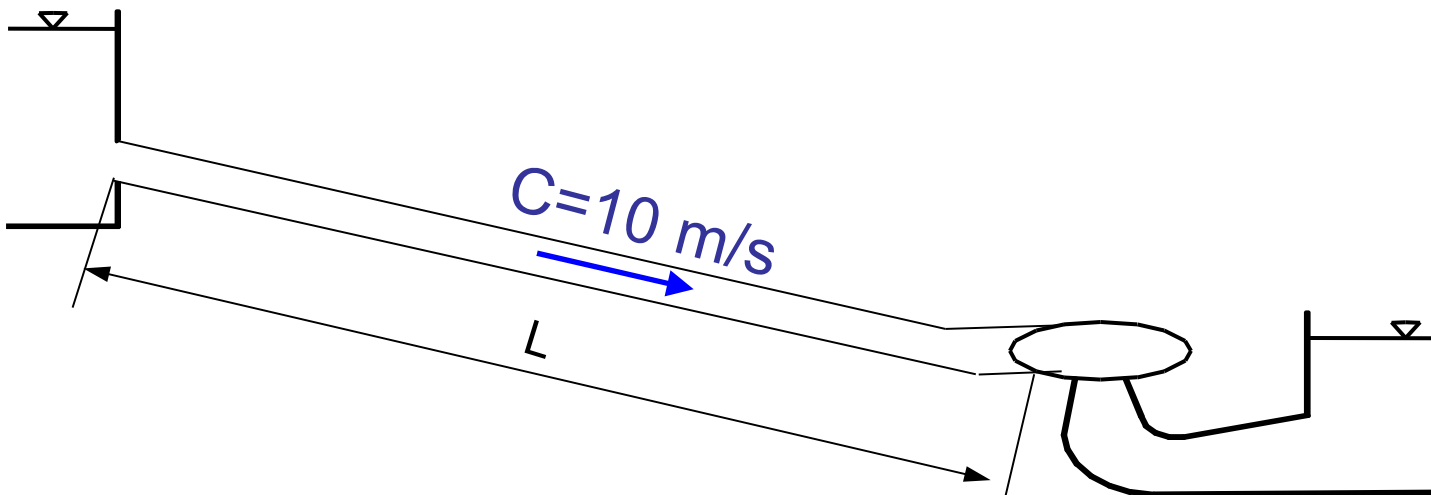
$$L = 300 \text{ [m]}$$

$$T_C = 10 \text{ [s]}$$

$$c_{\max} = 10 \text{ [m/s]}$$

$$g = 9,81 \text{ [m/s}^2\text{]}$$

$$\Delta h_{\text{wh}} = \frac{c_{\max} \cdot 2 \cdot L}{g \cdot T_C} = 61 \text{ m}$$



Calculation of the pipe thickness

$$L \cdot D_i \cdot p \cdot C_s = 2 \cdot \sigma_t \cdot L \cdot t$$

⇓

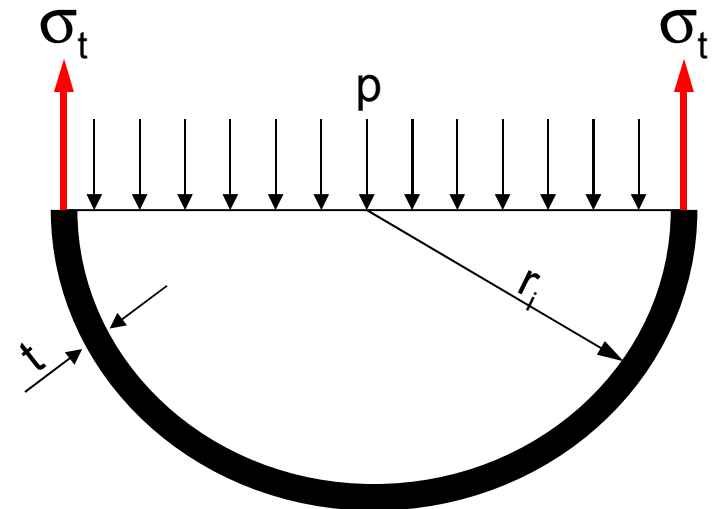
$$\sigma_t = \frac{p \cdot r_i \cdot C_s}{t}$$

$$p = \rho \cdot g \cdot (H_{gr} + h_{wh})$$

Where:

L	= Length of the pipe	[m]
D_i	= Inner diameter of the pipe	[m]
p	= Pressure inside the pipe	[Pa]
σ_t	= Stresses in the pipe material	[Pa]
t	= Thickness of the pipe	[m]
C_s	= Coefficient of safety	[-]
ρ	= Density of the water	[kg/m ³]
H_{gr}	= Gross Head	[m]
Δh_{wh}	= Pressure rise due to water hammer	[m]

- Based on:
 - Material properties
 - Pressure from:
 - Water hammer
 - Static head



Example

Calculation of the pipe thickness

$$L \cdot D_i \cdot p \cdot C_s = 2 \cdot \sigma_t \cdot L \cdot t$$



$$t = \frac{p \cdot r_i \cdot C_s}{\sigma_t} = 0,009 \text{ m}$$

$$p = \rho \cdot g \cdot (H_{gr} + h_{wh}) = 1,57 \text{ MPa}$$

- Based on:
 - Material properties
 - Pressure from:
 - Water hammer
 - Static head

Where:

$L = 0,001 \text{ m}$

$D_i = 2,0 \text{ m}$

$\sigma_t = 206 \text{ MPa}$

$\rho = 1000 \text{ kg/m}^3$

$C_s = 1,2$

$H_{gr} = 100 \text{ m}$

$\Delta h_{wh} = 61 \text{ m}$

Length of the pipe

Inner diameter of the pipe

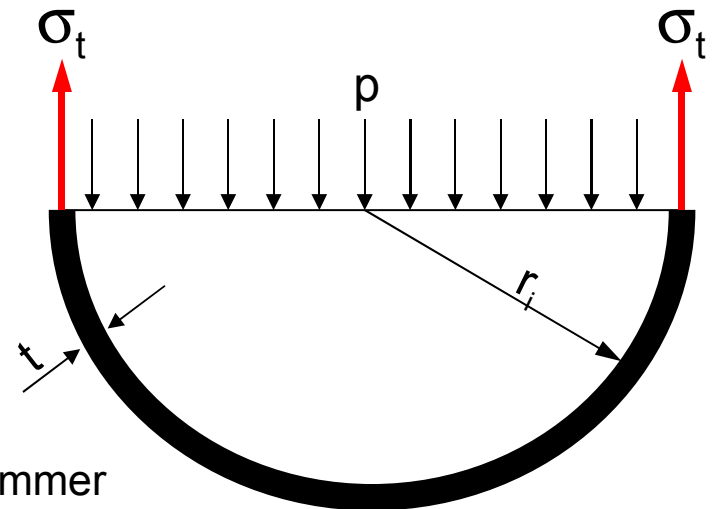
Stresses in the pipe material

Density of the water

Coefficient of safety

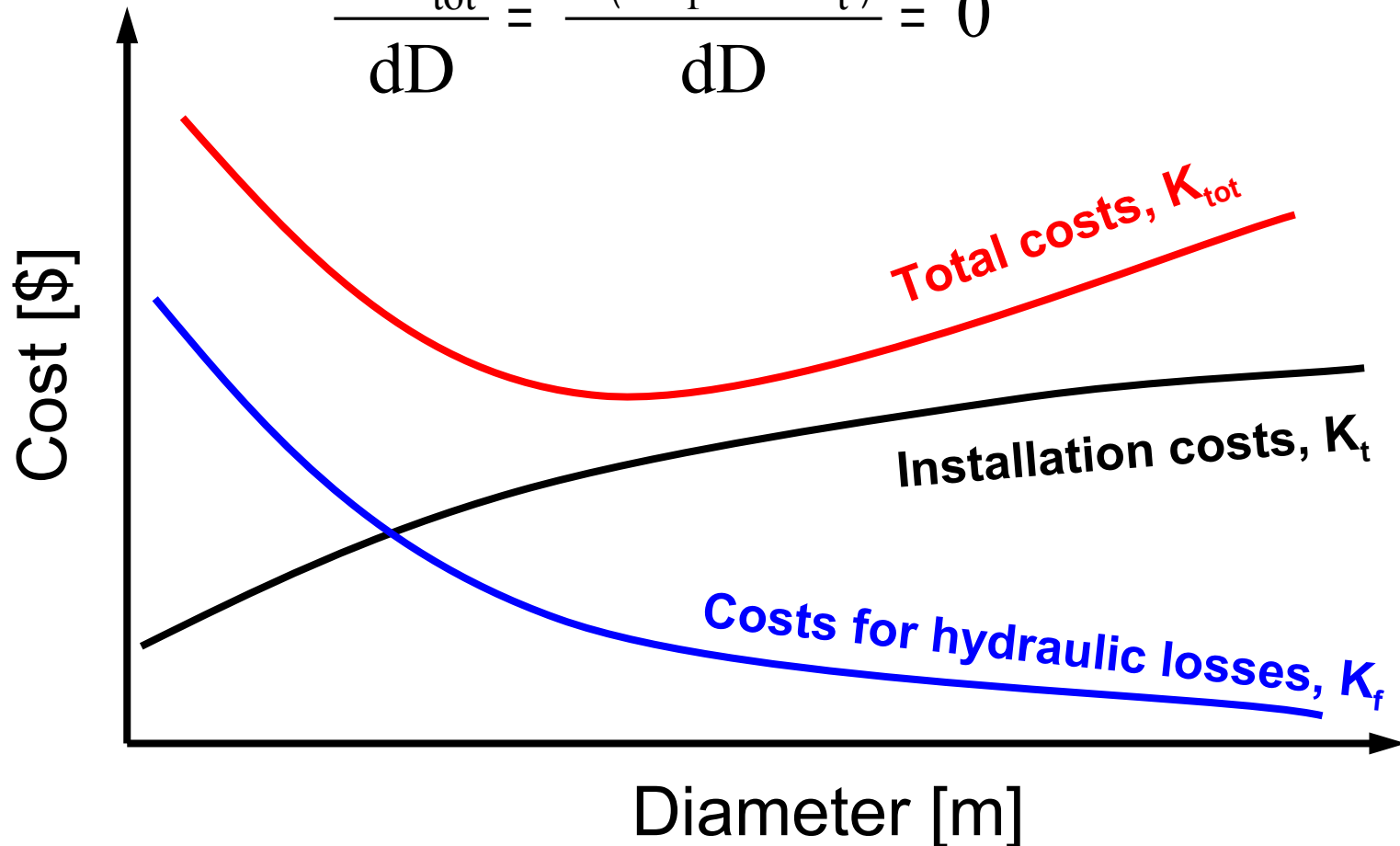
Gross Head

Pressure rise due to water hammer



Calculation of the economical correct diameter of the pipe

$$\frac{dK_{\text{tot}}}{dD} = \frac{d(K_f + K_t)}{dD} = 0$$



Example

Calculation of the economical correct diameter of the pipe Hydraulic Losses

Power Plant data:

H	=	100 m	Head
Q	=	10 m ³ /s	Flow Rate
η_{plant}	=	85 %	Plant efficiency
L	=	1000 m	Length of pipe

$$P_{\text{Loss}} = \rho \cdot g \cdot Q \cdot h_f = \rho \cdot g \cdot Q \cdot f \frac{L}{2 \cdot r} \cdot \frac{Q^2}{2 \cdot g \cdot \pi^2 \cdot r^4} = \frac{C_2}{r^5}$$

Where:

P_{Loss}	=	Loss of power due to the head loss	[W]
ρ	=	Density of the water	[kg/m ³]
g	=	gravity	[m/s ²]
Q	=	Flow rate	[m ³ /s]
h_f	=	Head loss	[m]
f	=	Friction factor	[-]
L	=	Length of pipe	[m]
r	=	Radius of the pipe	[m]
C_2	=	Calculation coefficient	

Example

Calculation of the economical correct diameter of the pipe
Cost of the Hydraulic Losses per year

$$K_f = P_{\text{Loss}} \cdot T \cdot \text{kWh}_{\text{price}} = \frac{C_2}{r^5} \cdot T \cdot \text{kWh}_{\text{price}}$$

Where:

K_f	= Cost for the hydraulic losses	[€]
P_{Loss}	= Loss of power due to the head loss	[W]
T	= Energy production time	[h/year]
$\text{kWh}_{\text{price}}$	= Energy price	[€/kWh]
r	= Radius of the pipe	[m]
C_2	= Calculation coefficient	

Example

Calculation of the economical correct diameter of the pipe
Present value of the Hydraulic Losses per year

$$K_f = \frac{C_2}{r^5} \cdot T \cdot kWh_{\text{price}}$$

Where:

K_f	= Cost for the hydraulic losses	[€]
T	= Energy production time	[h/year]
kWh_{price}	= Energy price	[€/kWh]
r	= Radius of the pipe	[m]
C_2	= Calculation coefficient	

Present value for 20 year of operation:

$$K_{f \text{ pv}} = \sum_{i=1}^n \frac{K_f}{(1 + I)^i}$$

Where:

$K_{f \text{ pv}}$	= Present value of the hydraulic losses	[€]
n	= Lifetime, (Number of year)	[-]
I	= Interest rate	[-]

Example

Calculation of the economical correct diameter of the pipe
Cost for the Pipe Material

$$m = \rho_m \cdot V = \rho_m \cdot 2 \cdot \pi \cdot r \cdot t \cdot L = \rho_m \cdot 2 \cdot \pi \cdot r \cdot \frac{p \cdot r}{\sigma} \cdot L = C_1 \cdot r^2$$

$$K_t = M \cdot m = M \cdot C_1 \cdot r^2$$

Where:

m	=	Mass of the pipe	[kg]
ρ_m	=	Density of the material	[kg/m ³]
V	=	Volume of material	[m ³]
r	=	Radius of pipe	[m]
L	=	Length of pipe	[m]
p	=	Pressure in the pipe	[MPa]
σ	=	Maximum stress	[MPa]
C_1	=	Calculation coefficient	
K_t	=	Installation costs	[€]
M	=	Cost for the material	[€/kg]

NB:

This is a simplification because no other component then the pipe is calculated

Example

Calculation of the economical correct diameter of the pipe

- Installation Costs:
 - Pipes
 - Maintenance
 - Interests
 - Etc.

Example

Calculation of the economical correct diameter of the pipe

$$K_{f\text{ pv}} = \sum_{i=1}^n \frac{\frac{C_2}{r^5} \cdot T \cdot \text{kWh}_{\text{price}}}{(1+I)^i} \qquad K_t = M \cdot C_1 \cdot r^2$$

$$\frac{d(K_t + K_f)}{dr} = 2 \cdot M \cdot C \cdot r - \frac{5}{r^6} \cdot \sum_{i=1}^n \frac{C_2 \cdot T \cdot \text{kWh}_{\text{price}}}{(1+I)^i} = 0$$

Where:

K_f	= Cost for the hydraulic losses	[€]
K_t	= Installation costs	[€]
T	= Energy production time	[h/year]
$\text{kWh}_{\text{price}}$	= Energy price	[€/kWh]
r	= Radius of the pipe	[m]
C_1	= Calculation coefficient	
C_2	= Calculation coefficient	
M	= Cost for the material	[€/kg]
n	= Lifetime, (Number of year)	[-]
I	= Interest rate	[-]

Example

Calculation of the economical correct diameter of the pipe

$$\frac{d(K_t + K_f)}{dr} = 2 \cdot M \cdot C \cdot r - \frac{5}{r^6} \cdot \sum_{i=1}^n \frac{C_2 \cdot T \cdot \text{kWh}_{\text{price}}}{(1 + I)^i} = 0$$

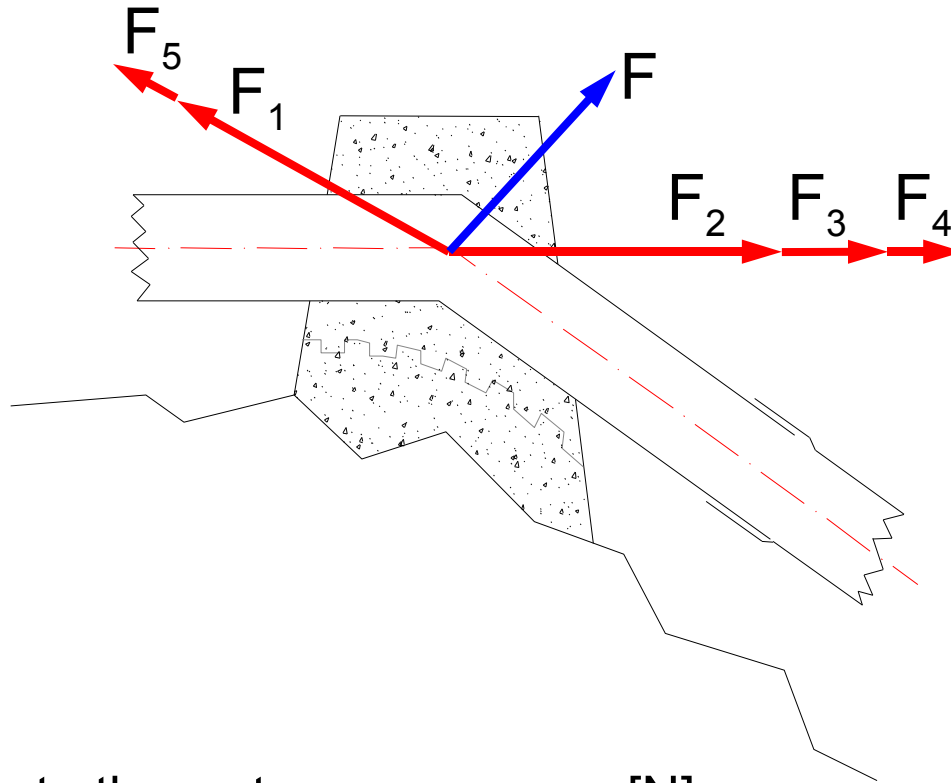
⇓

$$r = \sqrt[7]{\frac{5}{2} \cdot \sum_{i=1}^n \frac{C_2 \cdot T \cdot \text{kWh}_{\text{price}}}{M \cdot C \cdot (1 + I)^i}}$$

Calculation of the forces acting on the anchors



Calculation of the forces acting on the anchors



F_1 = Force due to the water pressure [N]

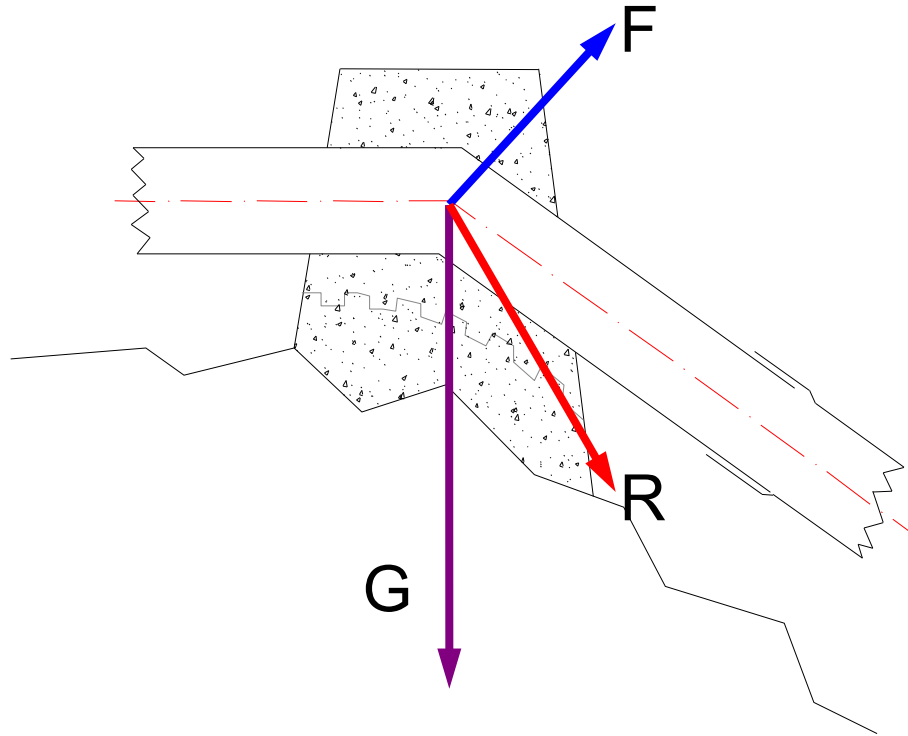
F_2 = Force due to the water pressure [N]

F_3 = Friction force due to the pillars upstream the anchor [N]

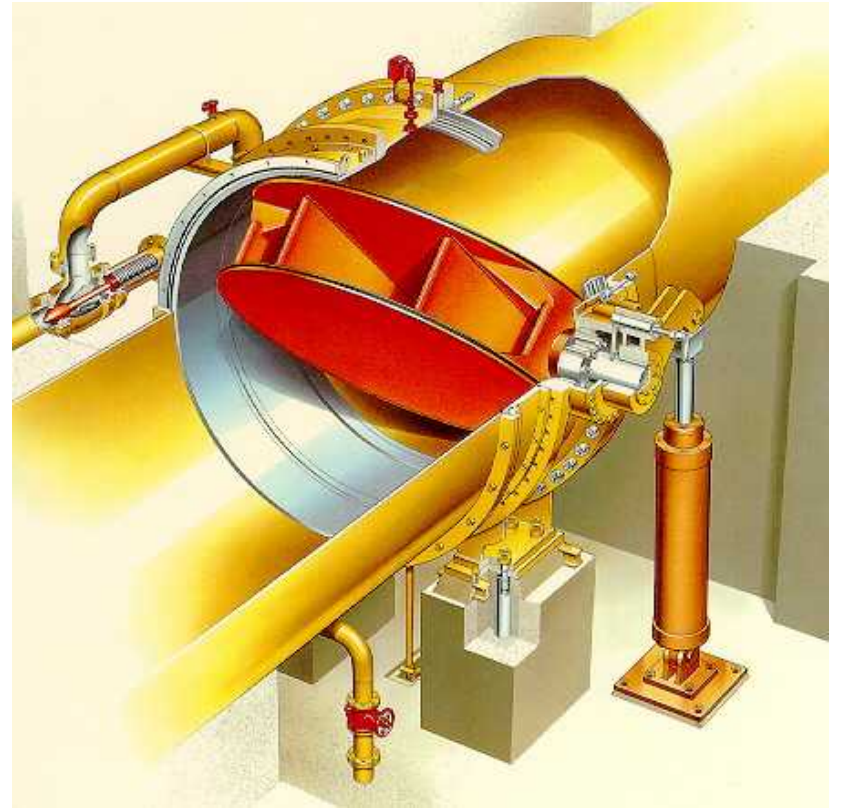
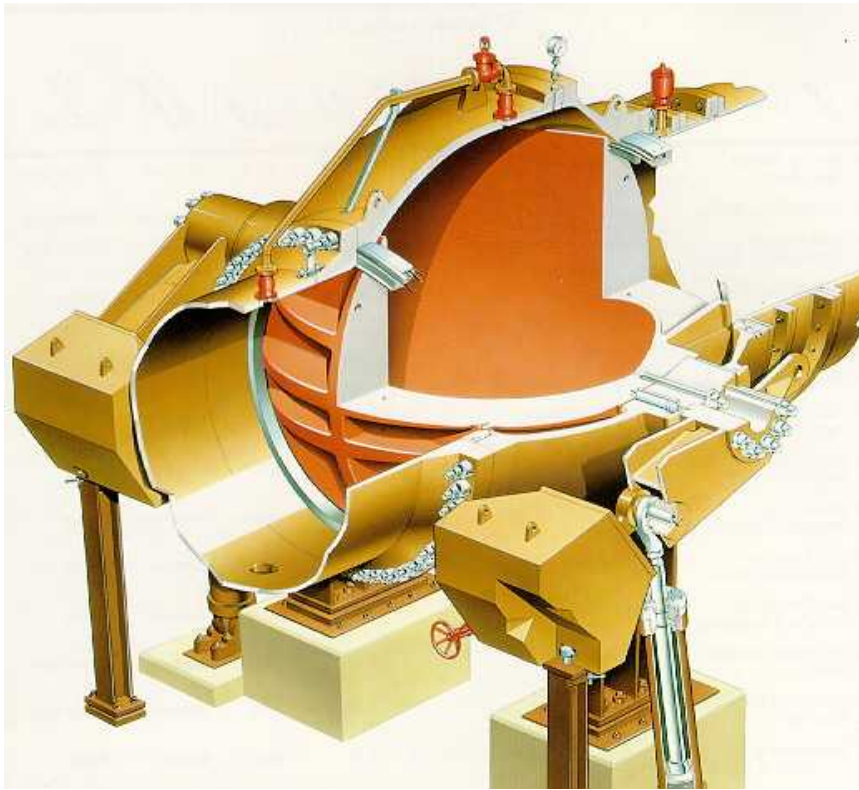
F_4 = Friction force due to the expansion joint upstream the anchor [N]

F_5 = Friction force due to the expansion joint downstream the anchor [N]

Calculation of the forces acting on the anchors

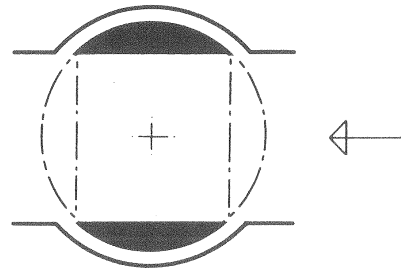


Valves

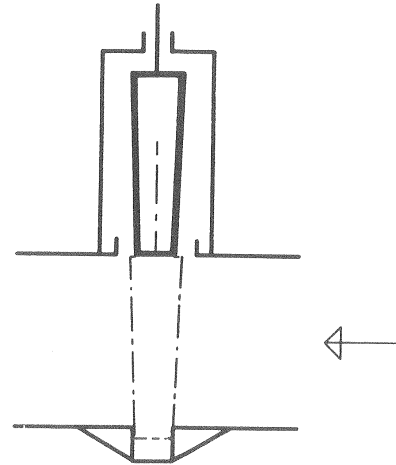


Principle drawings of valves

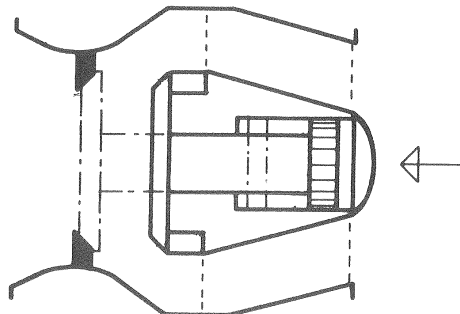
Open position ———
Closed position - - - - -



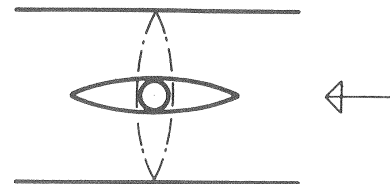
Spherical valve



Gate valve

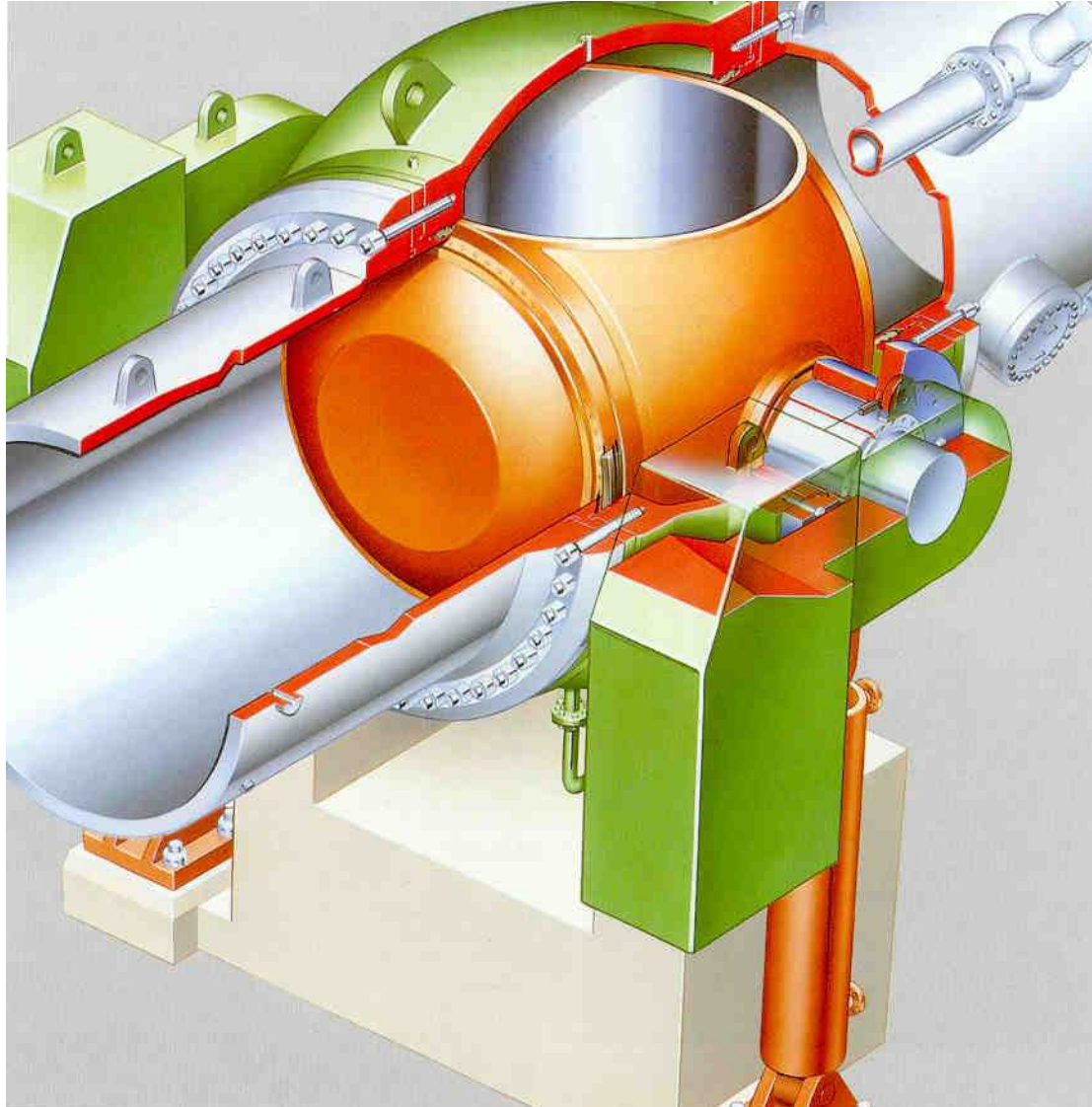


Hollow-jet valve

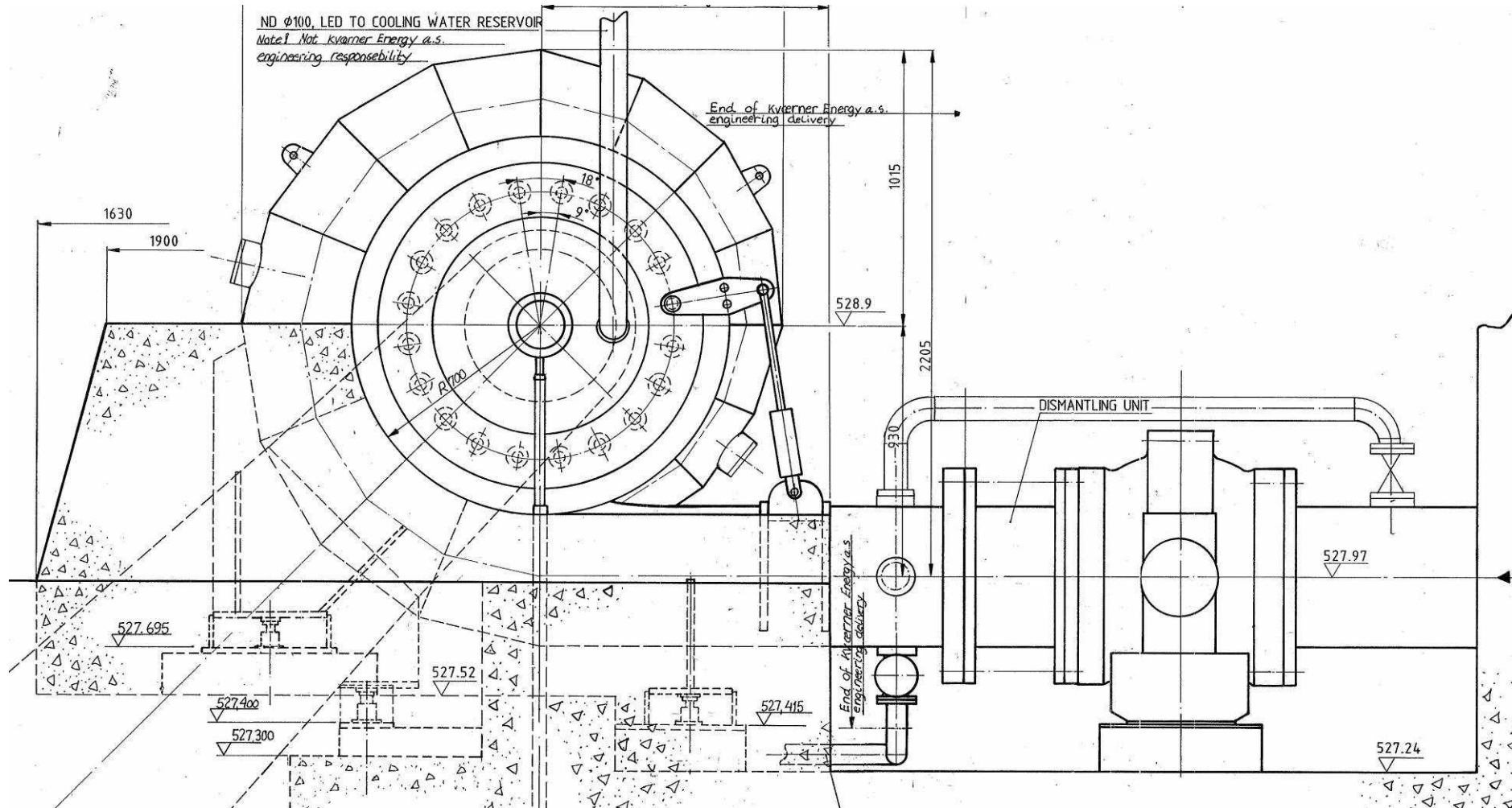


Butterfly valve

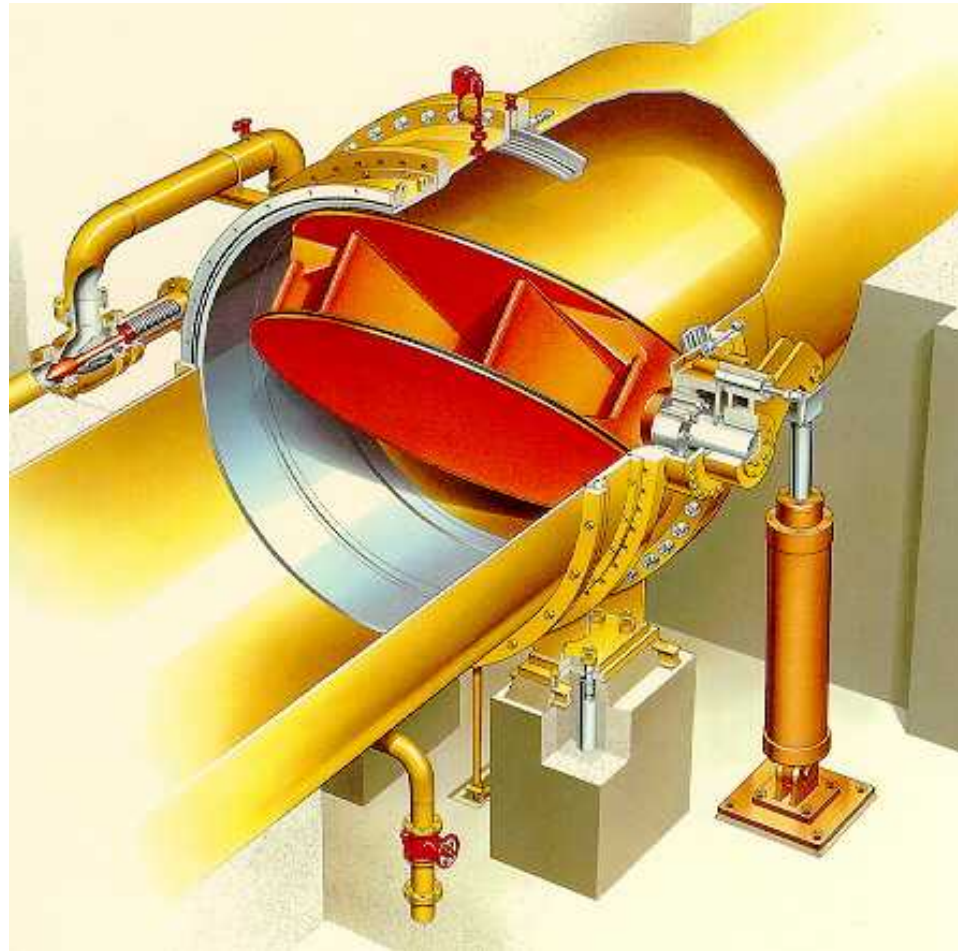
Spherical valve



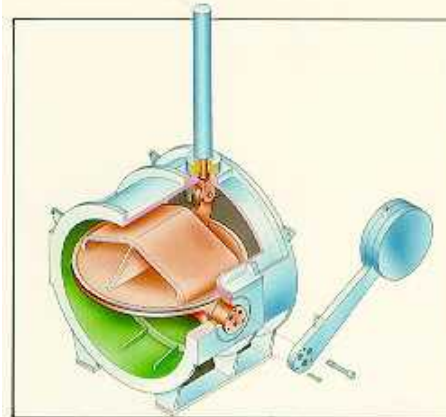
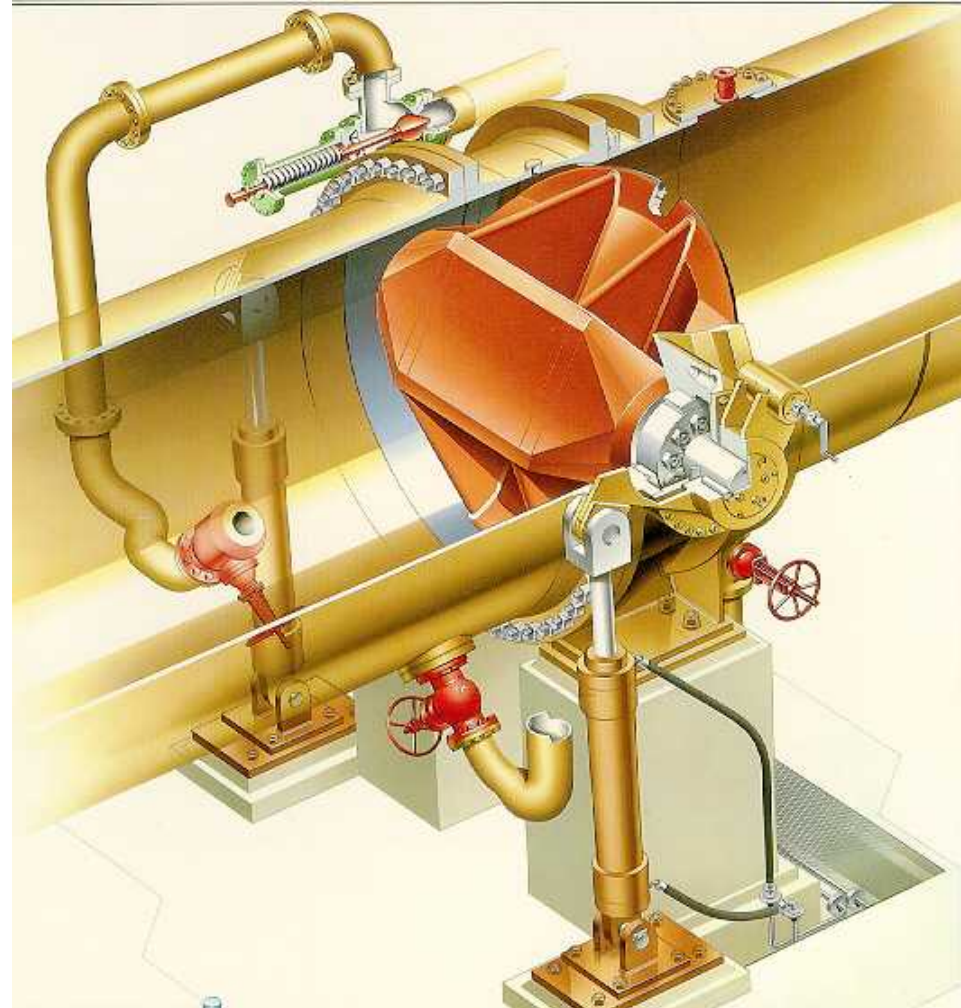
Bypass system



Butterfly valve



Butterfly valve

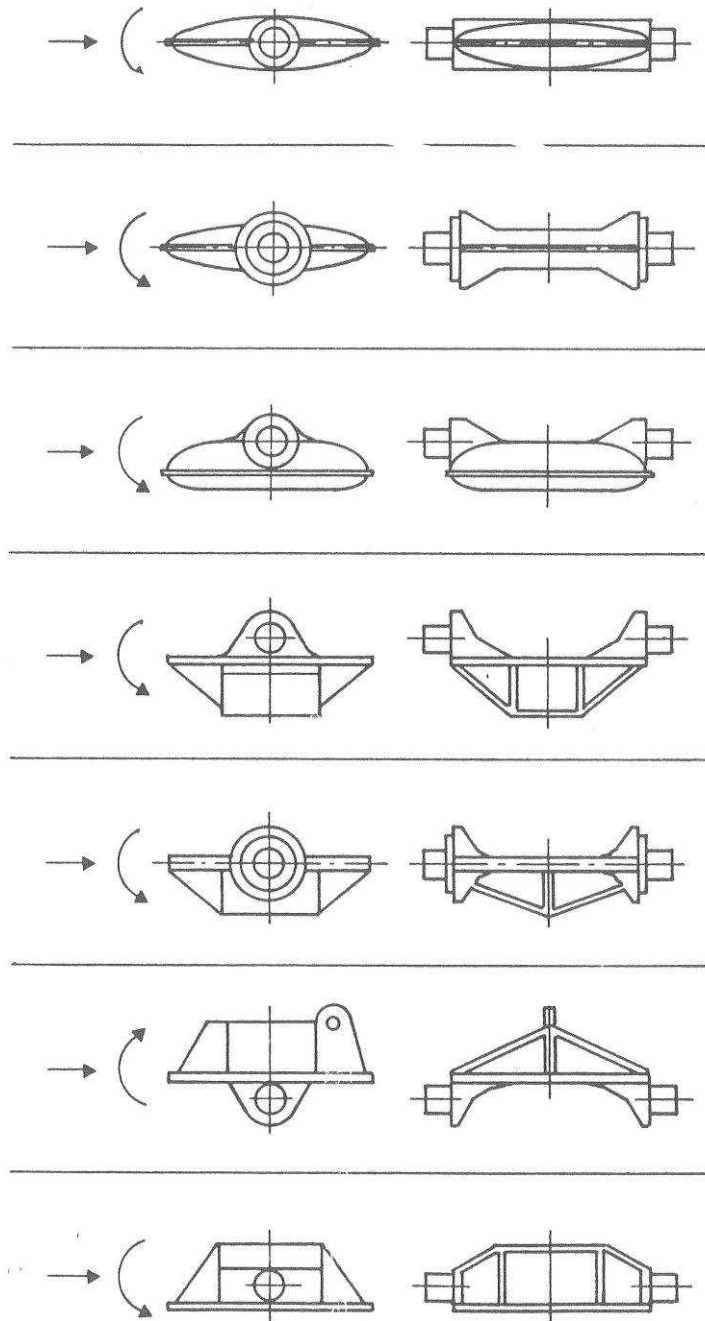


Lattice blade butterfly valve with blade mounted cylinder

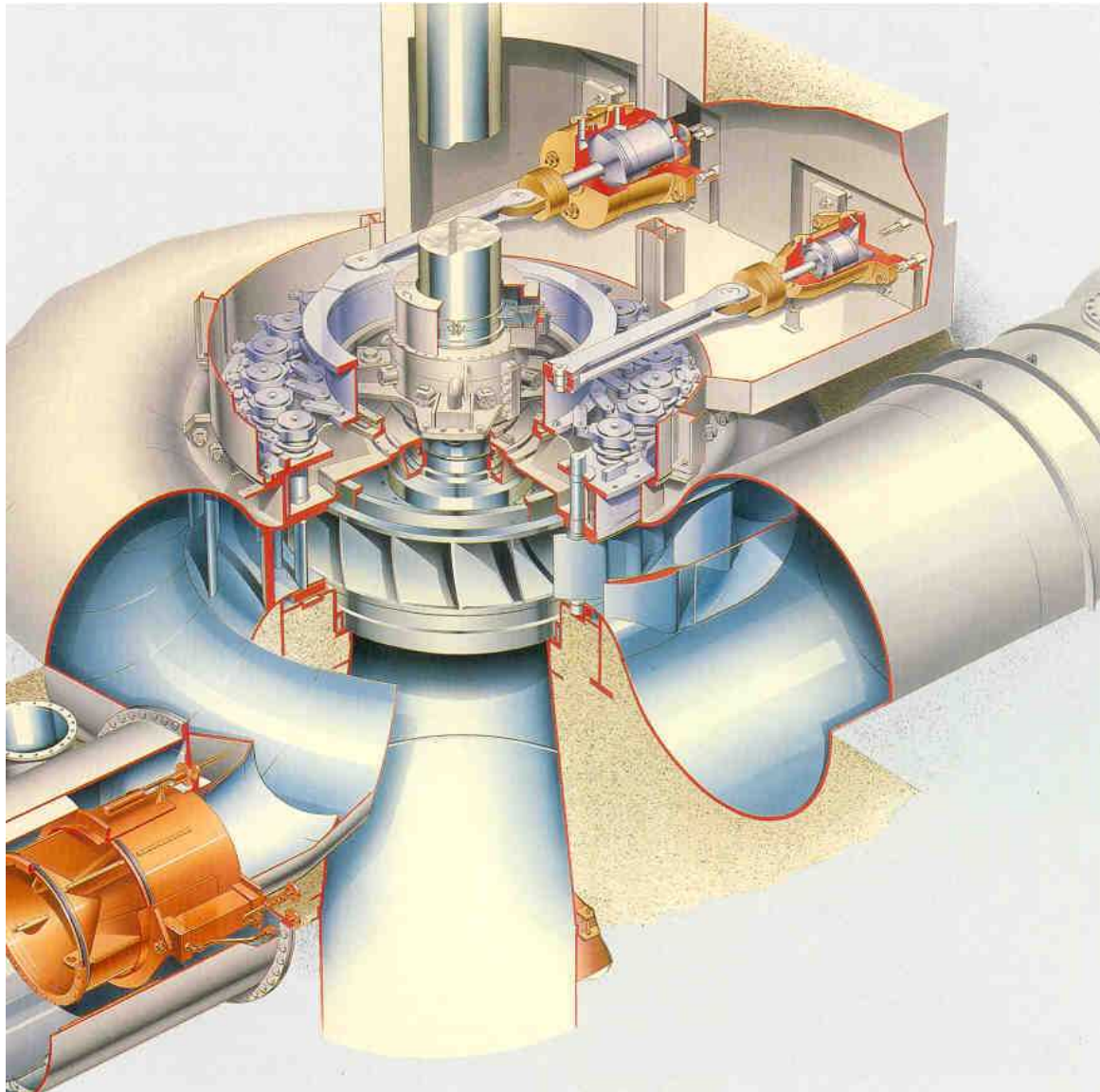


Turbine inlet valve, Steinskloss, Norway

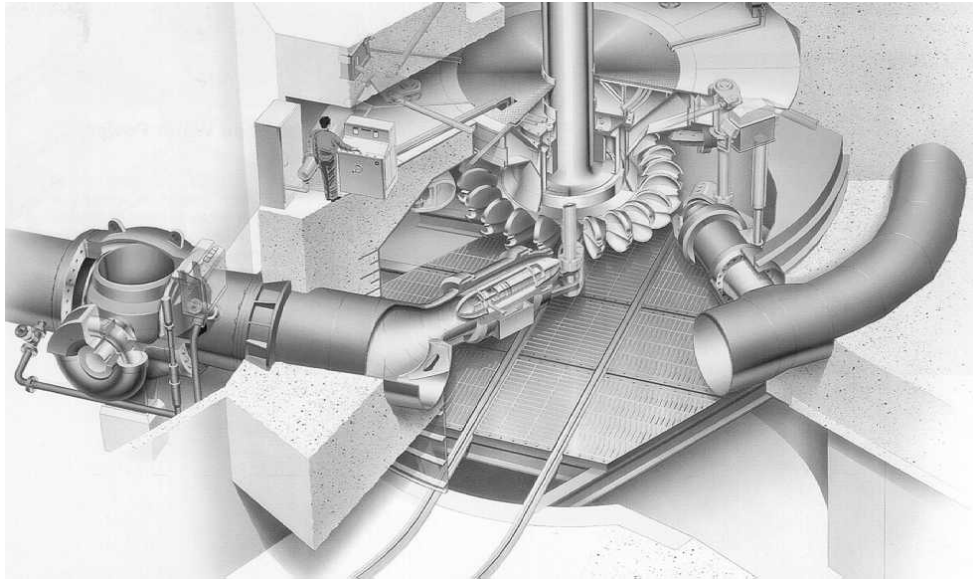
Butterfly valve disk types



Hollow-jet Valve



Pelton turbines



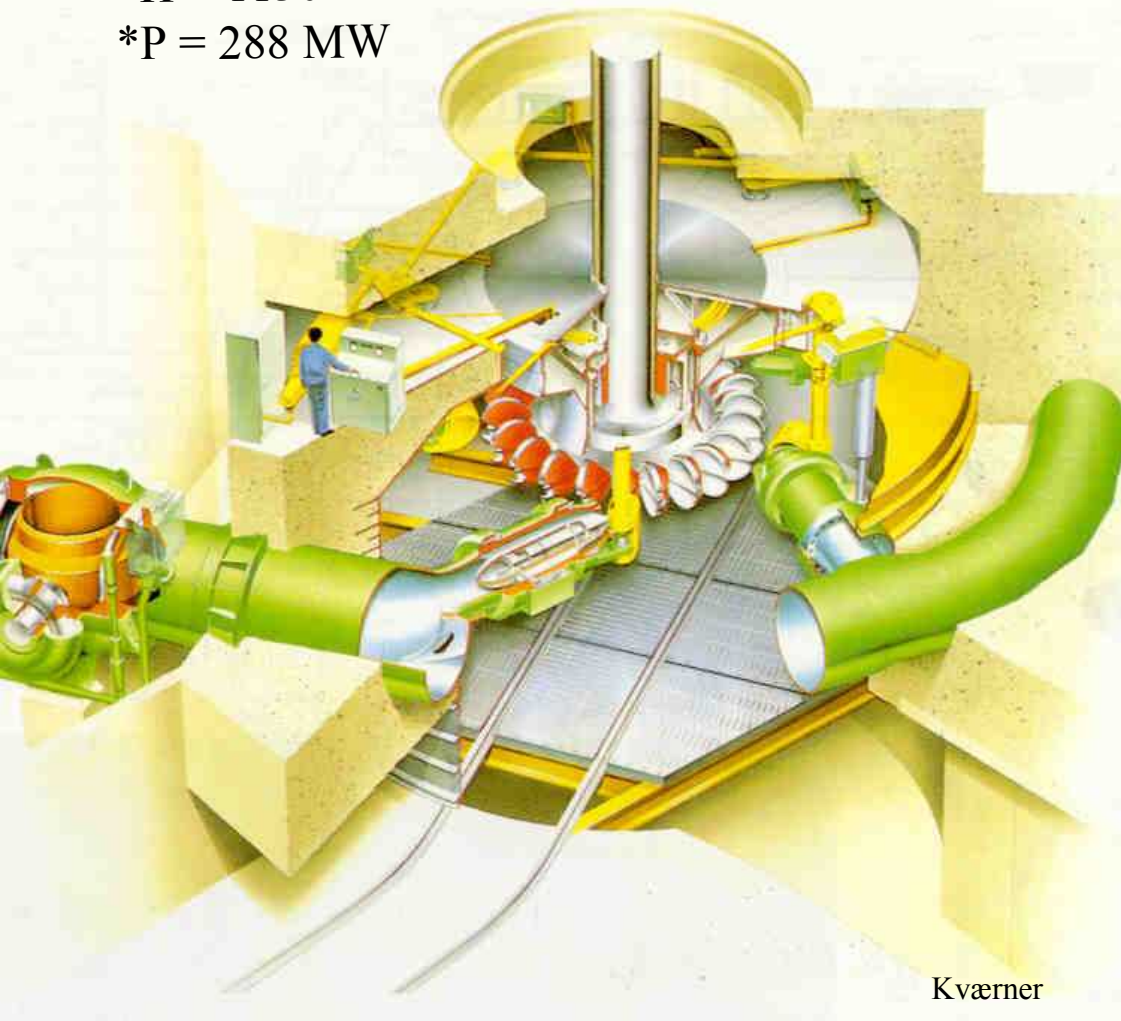
- Large heads (from 100 meter to 1800 meter)
- Relatively small flow rate
- Maximum of 6 nozzles
- Good efficiency over a wide range

Jostedal, Norway

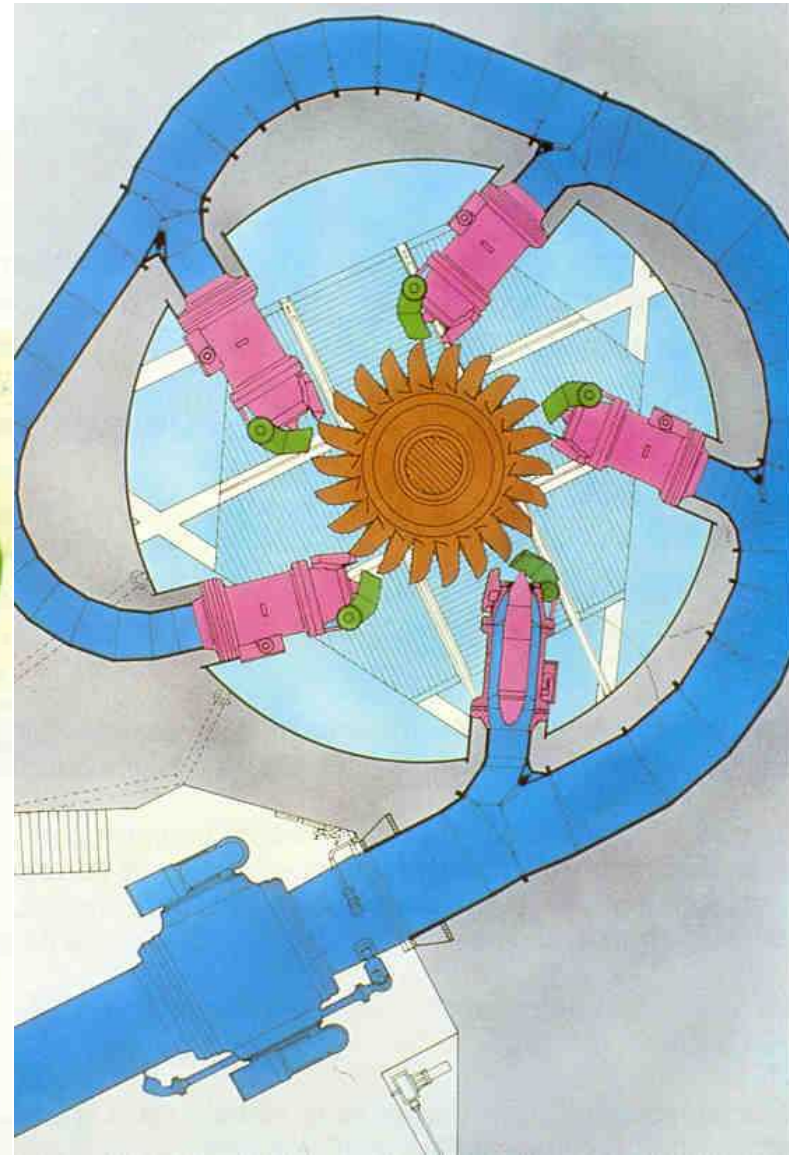
* $Q = 28,5 \text{ m}^3/\text{s}$

* $H = 1130 \text{ m}$

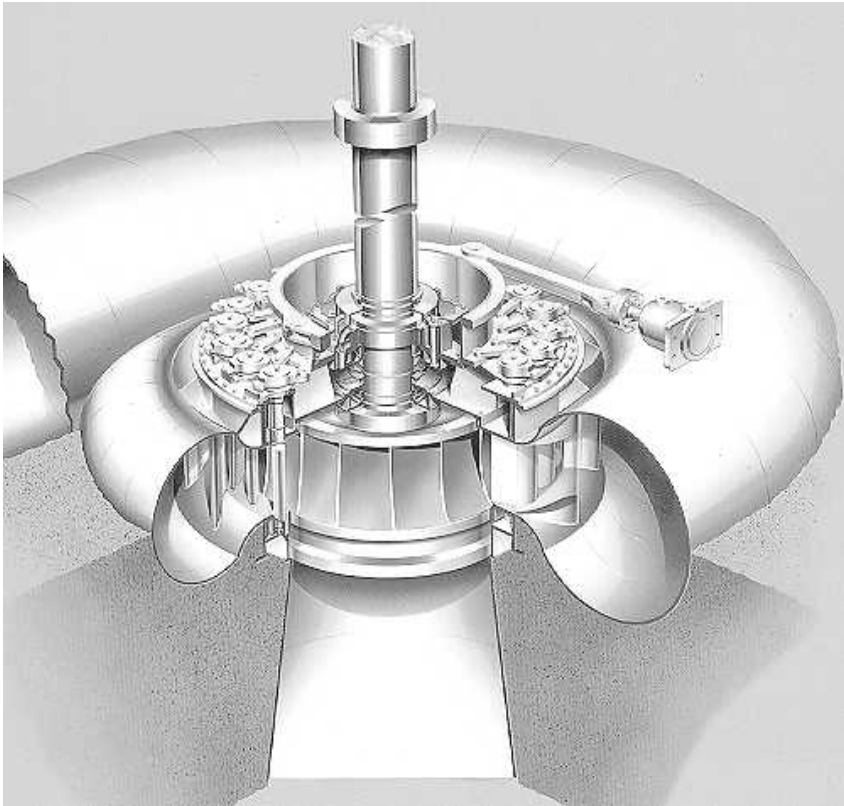
* $P = 288 \text{ MW}$



Kværner

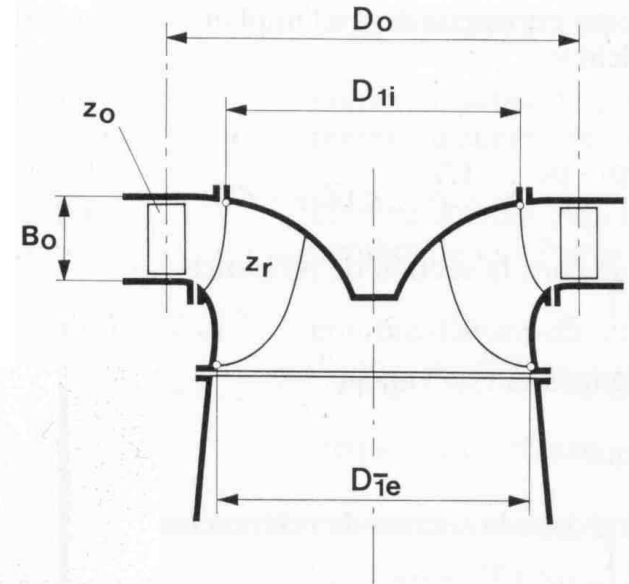
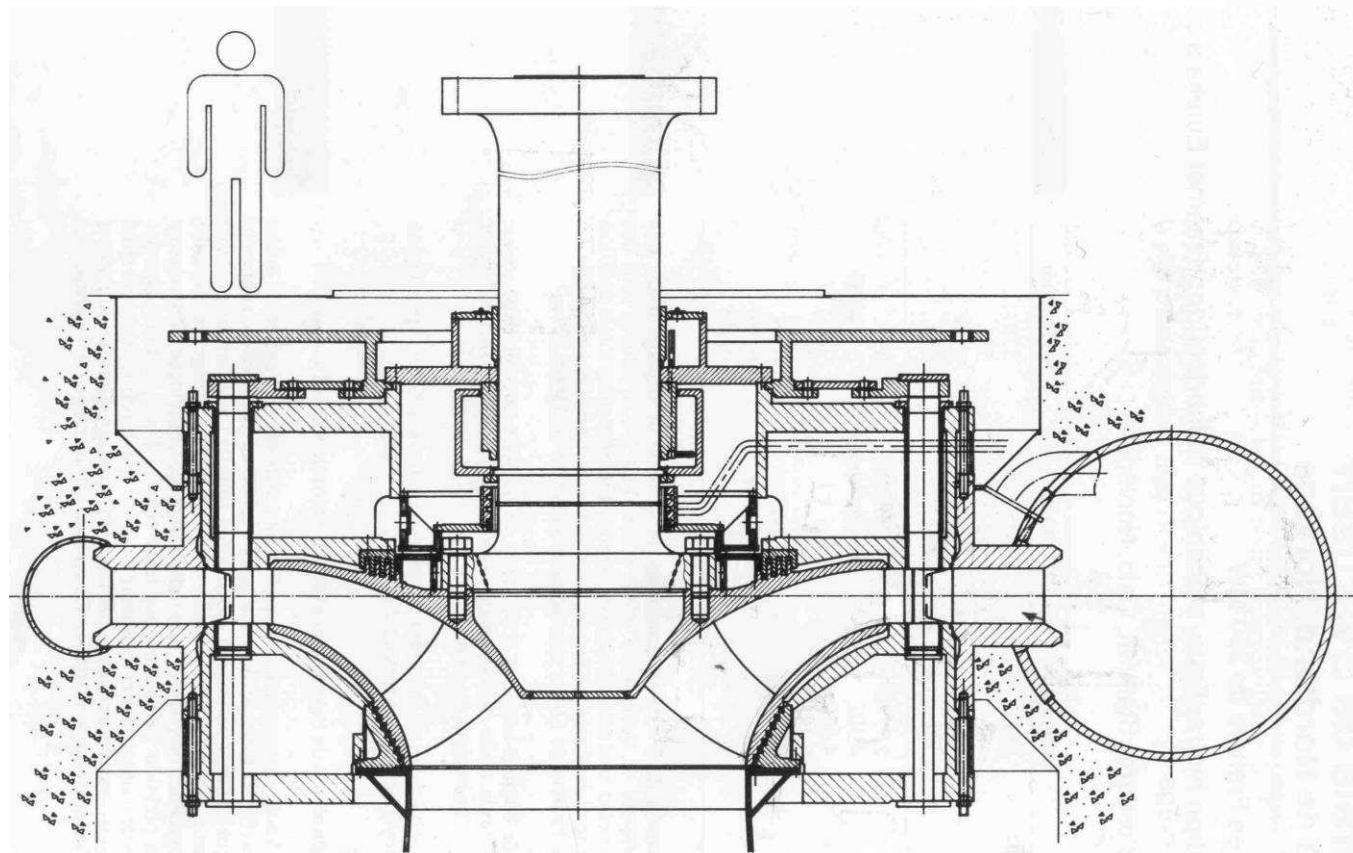


Francis turbines



- Heads between 15 and 700 meter
- Medium Flow Rates
- Good efficiency
 $\eta=0.96$ for modern machines

SVARTISEN



$$P = 350 \text{ MW}$$

$$H = 543 \text{ m}$$

$$Q^* = 71,5 \text{ m}^3/\text{S}$$

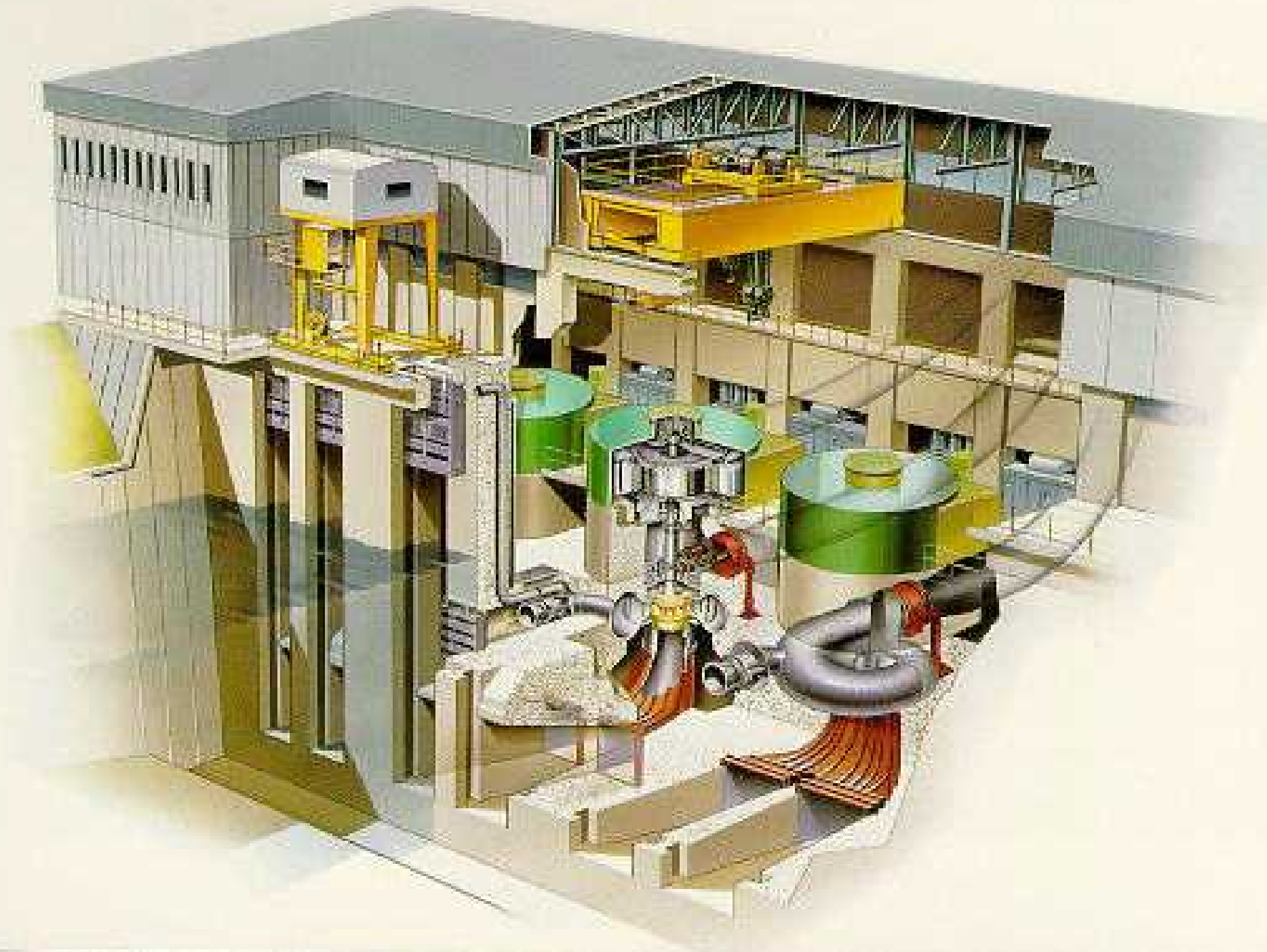
$$D_0 = 4,86 \text{ m}$$

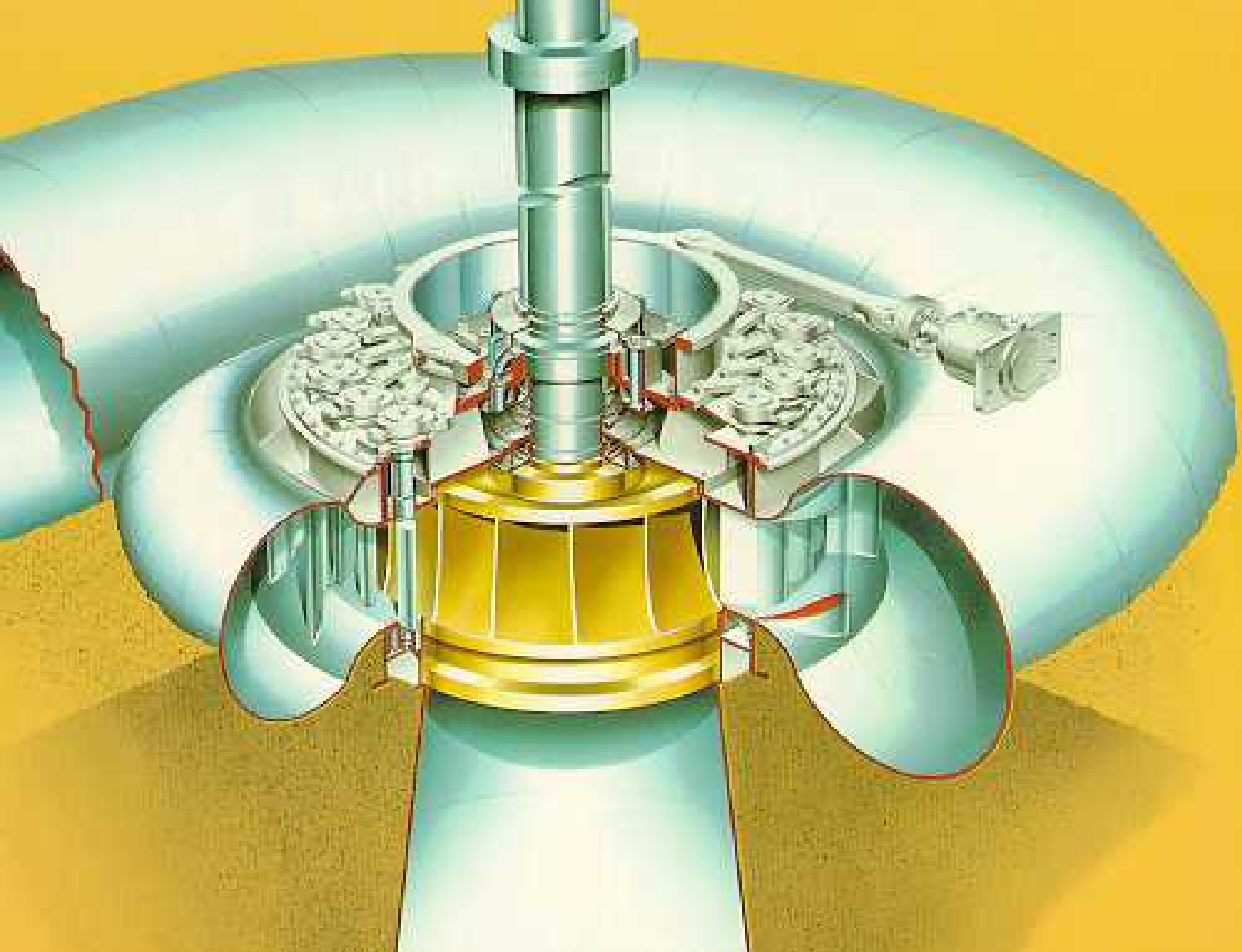
$$D_1 = 4,31 \text{ m}$$

$$D_2 = 2,35 \text{ m}$$

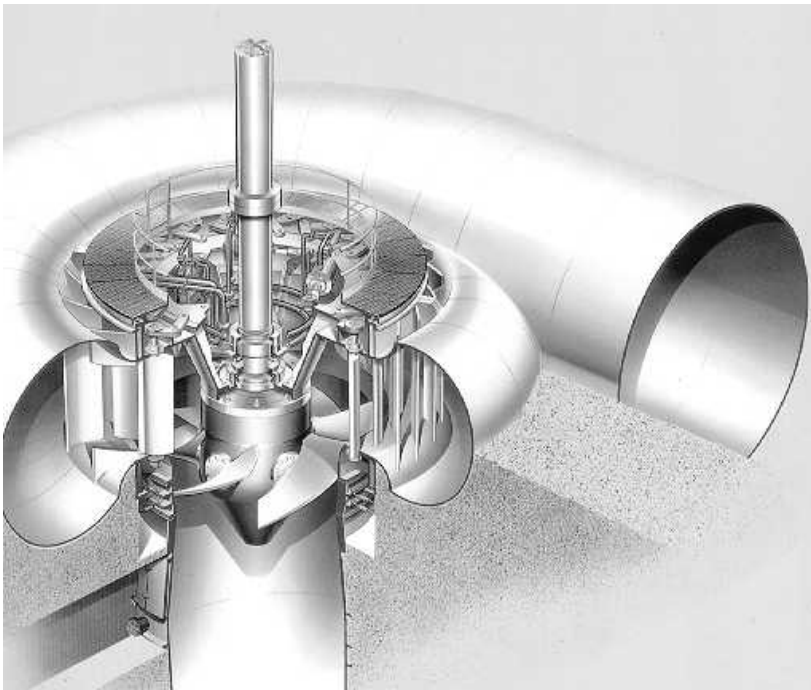
$$B_0 = 0,28 \text{ m}$$

$$n = 333 \text{ rpm}$$





Kaplan turbines



- Low head (from 70 meter and down to 5 meter)
- Large flow rates
- The runner vanes can be governed
- Good efficiency over a wide range

